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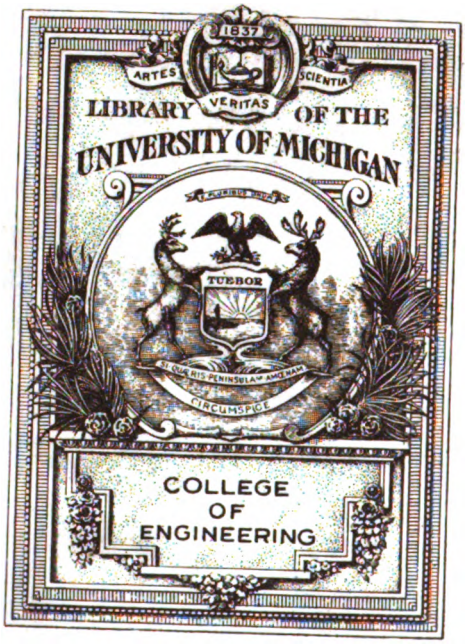

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INDEX TO AUTHORS

BUFFINGTON, H C—*Fundamentals of Tractor Engine Design*

A study of the number of cylinders from the standpoint of simplicity and operation for tractor use. The author's conclusion, based on the latest tests and in line with the most modern practice, can be briefly summarized as follows:

The four-cylinder engine is found to be the most adaptable and economical, in every way best suited to the work.

A two or three-bearing crankshaft (preferably the three-bearing) with a stress not to exceed 12,000 lb. per sq. in., and an explosion pressure of 300 lb. is recommended. The crankpin diameter of the bearings should be about 57 per cent of the cylinder bore and the length 1.2 of the crankpin diameter. The stress on the front bearing should not exceed 400 lb. per sq. in., on the center, 400 lb. and on the rear, 250 lb., taking the center of the bearings for point of reaction.

The removable head should have a liberal water jacket and even thickness of combustion chamber walls. The parting line of the head should be slightly below the piston. The stroke should be long, about 1.4 times the bore. In calculating the flywheel dimensions the horsepower should be doubled and allowance made for a 10 per cent variation in speed. The speed should be governed so as to be held within about 10 per cent above normal. The placing of the spark-plug should have careful attention.

On fuel not under 42 deg. Baumé and with open flash point not over 120 deg. fahr. a consumption of one pint per hp. hr. should be allowed. The engine rating on kerosene should not be over 75 per cent of the maximum capacity on gasoline, based on 800 ft. piston speed. Control of the heat used on the intake manifold and of water used with the fuel should be automatic.

Until a better system of using kerosene as fuel is employed, fresh oil should be supplied for lubrication to all moving parts and the used oil drained from the crankcase. Hand-holes should be made in the crankcase. Light base-pans should be arranged to be easily removed, and no small nuts should be used in out of the way places. Clearance should be provided underneath the engine on all sides. Tractor parts should not be fastened to the engine bolts, especially on the cylinder head. Flexible couplings should be provided between the clutches and the transmission shaft. 208

(Discussed by Barthel, Davis, Lewis, Lille and Slonneger)

CARLSON, G W—*Design of Military Truck Axles*

The War Department selected the worm gear type of rear axle for Class B axles because of production conditions.

The governing note for the new designs was sturdiness and simplicity. An illustration of the success of this attempt to reduce field repair work in all possible ways, is the Class B axles, particularly, which have three bearing sizes only on front and rear, the front inner bearing interchanging with the worm bearing, the rear wheel bearings being alike and interchanging with the different bearings, and the front wheel outer being the only odd size. The front carries only one size of screw, thirty-four of these being used. In the rear the only right and left-hand parts are the wheel-spindle units and the brake anchors. Brake levers, brake shafts, shaft brackets and toggles are all interchangeable.

The Class A and Class B front axles are, except for proportionate dimensions, the same. The regular Elliott type was chosen because of the opportunity it presented of bringing the pivot centers close to gage line, and because it is a simpler manufacturing proposition. So far as possible in this work, S. A. E. materials and heat treatments were used.

The knuckle pin, with oil lubrication, is an unusual feature.

Class AA front axle is of simple, passenger car type. Particular attention was paid to lubrication, which is with grease throughout.

The brakes are side by side and are made of rolled stock, toggle-operated, 24-in. diameter and 2½-in. wide. Since the War Department permitted a 24-in. diameter the problem was merely one of designing a brake easy of adjustment and of disassembling or relining. The toggle brake in this instance was given preference because it made accurate forming of bands and machining un-

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necessary. The objection raised on account of the rattle is a secondary item when the braking power available is considered.

All bearings are tapered roller except the internal-gear spur pinion and the wheel bearings on the Class A truck, which are of straight roller type. The manufacturers making the internal-gear axles recommended this practice on the ground that failure to adjust would cause a misalignment of the gears.

Oilers and grease cups have been omitted wherever possible and pipe plugs substituted. All nuts of large diameter and fine threads are case-hardened.

The internal-gear type was selected for the Class A rear axle, because of available manufacturing facilities and high ground-clearance. The countershaft is placed in the rear of the load-carrying member, which simplifies assembling and secures lighter weight. Class AA rear axle is of the full-floating passenger car type.

Many line drawings of unusual value illustrate the various points made by the author. For such special features as the brake system and the worm gear separate drawings were made. 125

CLARK, COL V E—*Types of Military Airplanes*

It will be America's part in the aircraft program to select from the many types developed by the Allies those that will best lend themselves to production on a large scale. Standardization means fewer types and greater numbers of machines.

Airplanes are needed for three purposes—observation, combat and pursuit, and destruction and harassment. Special types there will always be but these need not complicate the situation.

Bombing operations provide practically all the real damage it is possible for an airplane to inflict upon the enemy. Day and night bombers are differentiated and discussed as to details in type, limiting weight, powerplant and fuel, bomb capacity and minimum man-power.

The author sees in the slow-flying night bomber the means for a continuous and unrelenting attack on every point of strategical importance behind the lines of the enemy. Even consistent bombing of factory towns would end the war surely and quickly. 91

CLARK, W G—*Lubrication of Tractor Engines*

Many systems and combinations of systems are used in tractor engine lubrication. The author treats only of that part of the subject which has to do with the engine, dwelling especially upon the relation of lubrication to carburetion and ignition and upon the importance of proper selection of oil.

The method of oiling should be determined by the type of engine and its class of service. When apparently suitable oils have been selected they should be subjected to careful laboratory and field tests. An explanation of the cold, friction and power tests is given, the factors determining efficiency being clearly stated.

The author closes with these significant words: "The problem is one of cooperation among gas engine engineers, ignition and carbureter makers and oil refiners. Research and development must be carried out in conjunction with designing." 445

(Discussed by Buffington, Clark, Dasey, Fortner, La Schum, Mahoney and Plimpton)

COPLAND, A W—*Transmissions for the Class B Truck*

The Government asked for volunteers from the S. A. E. members to go to Washington and design two sizes of military trucks. The Class B truck was needed first and engineers were instructed to adopt one design only for each detail part to the last screw, bolt and nut. So much preliminary work had been done in previous meetings of the Transmission Division of the S. A. E. Standards Committee that the main features of the new design were quickly decided upon. The gears and the countershaft were practically the only parts not changed. The B transmission is of the amidships type with four speeds forward and one reverse. The main shaft and countershaft are placed in a horizontal plane with the idler gear-shaft above and to the left, looking from the rear forward. The horizontal type was adopted in order to get the ground clearance specified, 18 inches, below the transmission at a point midway between the front and rear wheels.

Although the units of the Class B truck were designed by separate groups, the different groups kept in close touch with each other and the result is a truck of balanced design. The Class B transmission

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is in no sense a compromise design. Each part of the design had practically the unanimous approval of the transmission engineers on the job, and in most cases represents successful, current practice. It is confidently expected to give a good account of itself under the most trying conditions on the battle front. Throughout the whole design S. A. E. standards were used just as completely as they were available.

The line drawings used make a valuable addition to the paper, although actual working dimensions are not given. To the drawings on control, however, some figures are added. 136

CORSE, W M—*Worm Gear Bronzes*

Interesting facts are given on the history of bronze and its relation to worm gearing together with references to the best work done on the mechanical side of the problem. The hour-glass and straight types of worm gearing are compared, special reference being made to the use of different bronzes.

The properties of phosphor bronze and aluminum bronze are given, and cooling curve diagrams of each alloy are shown. The chill castings of phosphor bronze are described and their advantages enumerated. Microphotographs of both phosphor bronze and aluminum bronze are shown. The special properties of aluminum bronze with reference to its use in worm gearing are given, and a table of physical properties of aluminum bronze is added. 234

DEAN, E W—*Fuel for Automotive Apparatus*

The paper deals with the relation between the industries of automotive engineering and petroleum refining, with particular reference to the fuel problem. The three general methods in use for the production of gasoline are discussed.

(1) Simple distillation from crude oil

(2) Separation of liquid fractions from natural gas

(3) Cracking of high-boiling petroleum products

Characteristic physical and chemical properties of the three types of products are mentioned and gasoline marketed at present are compared with those in use several years ago. It is shown that well-refined motor fuels of the newer types are of excellent quality, even though they lack some of the desirable properties of the "high test" gasolines formerly in general use. The change has been made necessary for purposes of economy and conservation of natural resources.

Probable future developments in the automotive fuel situation are discussed. Alcohol apparently will not be important until the more distant future. Coal-tar distillates, which probably will be utilized as motor fuel after the war, are limited in quantity. Oil shale is a promising although as yet undeveloped source of supply. Petroleum resources must be used to better advantage, even though new supplies of crude are discovered. Refiners must increase their yields of gasoline, and automotive engineers must design engines that will use fuels of a wider range of volatility. Cars should be produced with the idea of giving maximum service per unit quantity of fuel consumed, even if it prove necessary to sacrifice such qualities as luxury in use and maximum speed. 143

DOBLE, ABNER—*Electrically Controlled Steam Automobiles*

In the early nineties a wave of progress spread throughout the Continent, England and the United States, which started the development of the internal combustion engine as well as the electric motor for road vehicles. During the early development of automobiles even the long-established prestige of the stationary steam engine seemed to be somewhat in jeopardy. Much attention was given to the perfection of a gas engine for stationary plants. All this was natural yet from many points of view unfortunate, and the author claims that steam—the most flexible and most dependable source of power—should keep its place and popularity in the rapidly extending industrial possibilities of the power propelled vehicle. The results of ten years of experimental work are admirably condensed in the paper. In turn the steam generator, the ignition system, the construction of the engine and its lubrication are discussed. The advantages of steam power are clearly stated and a table of performance is added. The author then studies his

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subject from the point of view of the manufacturer, under the war-development idea of quantity production. 338

(Discussed by Brown, Castle, Emory, Gibson, Schlesinger, Sturgess, Thayer, Tillson and Warren)

DUNHAM, GEORGE W—*Presidential Address* 28

GOLDBERGER, ERNEST—*Economic Size of Farm Tractor*

A comparative analysis of the cost of plowing with small, medium and large size tractors is presented and plotted on charts. This analysis is based upon the following terms: Purchasing cost, $600 for small, $1500 for medium, $3,600 for large size tractor; one operator for either small or medium, two operators for the large size tractor; same fuel cost per hp.-hr.; complete depreciation of machines after 4500 hours of plowing; 6 per cent on the undepreciated value of the outfit for interest and repairs.

The chart showing the plowing cost in dollars per acre for any number of acres plowed per year, leads to the following conclusions:

(1) The cost of plowing increases rapidly with decreasing acreage.

(2) The four-plow tractor is more economical than the two-plow where more than 240 acres have to be plowed per year, six inches deep, or more than 180 acres per year, eight inches deep.

(3) One or two four-plow tractors are more economical than one eight-plow tractor under any conditions of use.

If the proportion in the purchasing price of two to four-plow tractors is greater than the one assumed (600:1500), the four-plow tractor figures out the more economical even on farms with much less than 240 acres of plowed land.

Furthermore, the advantage of completing the plowing early and quickly, and ability to do all the belt work on the farm, the modern movement toward intensive farming and deeper plowing, the increasing difficulties in the competition between the very small grain farm and the larger and more economical farm unit, all indicate that the four-plow tractor is the probable standard-size farm tractor for the near future. 397

GREER, E R—*Tractor Power System*

A tractor power system includes all the power transmitting parts from the engine to the ground, and to a pulley for belt work. There is very little uniformity and much difference of opinion among engineers as to what constitutes correct practice. Attention is called in the paper to important items, such as traction members, the type of final drive, distribution of weight, protection of parts and bearings, and this is followed by a discussion of each gear reduction, in turn. The transmission as a whole is a series of compromises, each part being affected by the design of the other parts. 220

(Discussed by Davis and Trask)

GURNEY, F W—*Some Fundamentals of Rolling Support*

The essential thing in rolling actions is stationary contact—no slippage. Movement should take place only where there is no contact.

The basic elements of bearings, either ball or roller, are the outer and inner race rings and the rolling elements, which may be either balls or various kinds of rollers. The paper reviews the characteristics of these different types, and the advantages and disadvantages of each. As rollers have an apparent advantage in "line contact" over the "point contact" of balls, the validity of the "line versus point contact" argument is examined, and the alleged weakness of point contact discussed. With proper raceways the term "point contact" is a misnomer. According to the author raceways can be made to give ample areas of contact to carry any loads that may be imposed.

The difficulties incident to misalignment, imperfect contact and slippage are discussed, as slippage under metal-to-metal contact causes great friction. Alignment difficulties however do not apply to ball bearings, since there is neither alignment nor misalignment to a ball. The remarkable accuracy and physical perfection of balls are due to the process of quantity manufacture, the sphere being self-determined.

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The advantages realized in load support by various degrees of race curvature are explained. It is said that by close race contour the ball bearing can be made the equivalent of the roller bearing in contact area, and this without incurring the embarrassments incident to alignment and misalignment. 293

(Discussed by Crawford, Gallup, McComb, Moskovics, Ricker and Smith)

ILLMER, LOUIS—*Heat-Flow Through Cylinder Walls*

A research study into the speed limitations imposed upon high-duty internal-combustion engines by excessive heat-flow through the cylinder walls.

Excessive heat-flow raises the temperature of the bore and head walls unduly, and this, in turn, leads to lubricating and cracking troubles. It is shown that large bore engines reach their limiting bore temperatures at comparatively low rotative speeds and that large high-speed engines are likely to suffer seriously from destructive heat-flow effects.

The determining factors which control heat-flow are shown to be: (a) temperature, (b) rotative speed, and (c) relation of volume to cooling surface. Formulas are presented which aid in determining heat-flow quantitatively, and a method is developed for analyzing critically the design characteristics of any normal gas or oil engine for heat-flow effects.

From a study of speed limits taken of numerous engines in various kinds of commercial service, the desirable bore-temperature limit is found to lie between 400 and 450 deg. fahr. The maintenance of the same bore temperature in all sizes of engines necessitates a reduction in the heat-flow rate with increased bore dimensions. With this as a basis, the maximum rotative speed can be rationally predetermined. The most suitable, "safe," speed limit for any given kind of engine is dependent upon the character of the service for which it is intended. For continuous heavy-duty demands, the stipulated speed limits must be reduced by a certain multiplying factor to provide a suitable margin of safety against excess wear and fatigue of the cylinder parts.

The combined temperature and pressure stress set up in a cylinder wall is also analyzed and formulas deduced for estimating it. The proper thickness of cylinder wall, such as will best insure against cracking, is likewise given.

The author points out that the heat-flow rate through the bore wall can be materially reduced by the use of a relatively large cooling surface per unit of volume. Such ample cooling surface provides for a better distribution of the jacket heat offtake and is especially effective in the case of large bore engines, which are inherently restricted in this respect. The skilful manipulation of this factor should lead to further improvement in internal-combustion engine design and allow large engines to run at piston speeds heretofore deemed prohibitive. 162

LANDIS, MARK H—*Shock Absorbers for Easy Riding*

The paper deals with shock absorbers used with automobile spring suspensions to secure easy riding.

The general design of spring suspension, considered first, covers the following factors: flexibility of spring mountings; weight of car; percentage of unsprung weight; spring rates; range of spring movements; interleaf spring friction; range of movement of front spring and control of spring recoils.

Shock absorbers are described and illustrated under three general classes, auxiliary spring, compression, and recoil damping. The various designs in each class are criticized, with a view to showing their limitations and possibilities.

The conclusions as reached by the author seem to demand a soft, flexible spring suspension with a large range of spring movement and with the hydraulic recoil damping type of shock absorber. 474

(Discussed by Blackledge, Carter, Farnham, Flentje, Liebau, Waterman and Watson)

MILBRATH, A F—*Design of Class B Truck Engine*

The author describes the designing of the Class B military truck engine by the corps of engineers assigned to this work, and shows in detail the development of the plans.

It was required to develop an engine with a torque output of at least 2800 in-lb., which was necessary to give the tractive factor

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pre-determined for this truck. The normal speed of 1050 r.p.m. had also been fixed, as well as an approximate speed of 1500 r.p.m. for maximum horsepower. The prime requisitions for this engine were liberal bearing surfaces and ample proportions in all parts, in order that repairs would be needed at infrequent intervals only, even under the severe conditions imposed by military service.

With these limitations the piston displacement, stroke-bore ratio, valve sizes, compression ratio and other proportions were worked out. The crankshaft stresses and bearing pressure were carefully calculated. The oiling system was specially designed to meet this service. Provision was made for draining the sediment frequently without drawing off all the oil.

Performance curves made from tests of one of these engines are given; these show that the results desired were more than realized in the completed engine. 115

MODINE, ARTHUR B—*Principles of Tractor Engine Cooling*

Many factors other than radiator size are involved in engine cooling, and the author discusses in turn the radiator, air velocity and fan power requirements. His explanation of the radiator covers heat to and from, heat equivalent in horsepower, exhaust loss and radiation and convection losses. Average tractor engine economy in the use of gasoline is one factor in the problem.

The author gives in conclusion a formula for radiator sizes and recommendations for water circulation, air velocity and fan details. 435

MOWRY, J L—*Fuels for Tractor Engines*

The present demand for a low-grade fuel is making serious trouble for both manufacturers and users. The fact that kerosene is not an engine fuel, in no way affects the determination to use it. There are more users of kerosene than there are burners for this troublesome fuel. Only 12 per cent of the tractor users in Minnesota burn gasoline.

The results of laboratory tests are presented, showing that the hot-spot manifold is more promising than other types.

The farmer is willing to pay for a good machine, but expects it to burn the fuel for which it is advertised. He wants also a light machine, but at present he cannot get one that will burn kerosene.

Of all possible future fuels for tractors, the fixed gas is the most promising. 464

(Discussed by Andrews, Bennett, Clark, Couse, Crandall, Greer and Whitney)

MOYER, AMOS F—*Rolling Resistance of Tractor Wheels*

The work done in rolling a wheel a given distance must be equal to that expended in depressing the soil over the same distance. A bed of freshly tilled soil in which conditions could be duplicated for repeated tests, and a stationary dynamometer to measure the pull required to move loaded wheels by means of ropes, prepared by the experimental engineering department of the University of Minnesota, formed the basis of the detailed studies and experiments which the author summarizes in his paper. It was soon discovered in these tests that velocity was the disturbing factor in proving the equations, and some very interesting curves are shown to illustrate this point. The author feels that proof of the laws which exist between the quantities involved will be of the utmost assistance when supplemented by actual field data. 405

(Discussed by Morelius, Scarratt and Seaton)

MYERS, CORNELIUS T—*Chassis Design of Class B Motor Trucks*

The paper gives a short résumé of the preliminary work done by members of the Society of Automotive Engineers on the Class A and Class B standardized Liberty motor truck chassis. The design of the Class B chassis is then explained in detail, and a comparison made between the specifications as first drawn and those finally embodied in the approved design.

The general design and the guiding factors in the accepted design are given, together with a short description of the principal features and the methods of mounting the various units in the chassis. A plan and side elevation of the chassis are given also, and analysis diagrams showing the interrelation of the various quantities in the author's "tractive factor" formula as applied to this

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particular chassis. The effect of modifications of these quantities on the performance of the truck is explained, as are figures actually applied to the truck. 104

OETTING, O W A—*Effects of Cold Temperatures on Starting*

A summary of tests made of the starting of different automobile engines to determine approximately the power required to crank any size of engine at a temperature of 10 deg. fahr. The results are given in terms of the piston displacement of the engine.

The effects of low temperature on starting engines are numerous and so variable that it would be difficult to make any theoretical calculation to determine them. The data given in this paper were secured by testing in a refrigerator at the desired temperatures in such a way as to approximate cold weather conditions.

The minimum size of storage battery for the satisfactory cranking of automobile engines is given. The illustrations show the sizes of battery in terms of piston displacement on various sizes of engines and also as determined by the results of these investigations. 243

(Discussed by Cook, McGéorge, Scaife and Strickland)

SCARRATT, A W—*Gears for Tractors*

The service demanded of tractor transmissions is continuous and unusually severe as compared with that demanded of automobile transmissions. For this reason careful judgment, good engineering, high-class material, accuracy and the best of workmanship are absolutely necessary, if service, light weight and maximum efficiency are to be insured.

Gears should be made of high-grade material and be accurately machined and carefully heat-treated in order to obtain maximum values of physical properties. They should also operate continuously in oil in dirt-tight housings.

Cast iron gears should be eliminated and a 20-deg. stub tooth, similar to that outlined by Charles H. Logue, adopted as standard. Gear faces should be of ample width and proper hardness to insure safe surface, pressures and reasonable wear.

The allowable stress curve of the Lewis formula should be replotted to make it more closely applicable to high-grade, heat-treated steels. The value of factor (Y) in the Lewis Formula should be revised to give true value for a standard 20-deg. stub tooth.

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(Discussed by Andrews, Buffington, Greer, Hatfield, Krieg, McDonough, Moyer, Nead and Young)

SCHELL, C A—*Flexible Fabric Universal-Joint*

Under the headings, life of flexible fabric joints, use with Hotchkiss drive, class AA military trucks, propeller-shaft joints, lubrication, bolts, disks and diameters, the author gives information concerning a universal-joint made up of fabric disks.

The assembling and manufacture of a fabric joint disk is illustrated and tests and examples are given. Experiments with different types of washers are described and comparisons made between three and four-arm types of spiders. Internal flange types of coupling are touched on and some figures given on the service obtained from pioneer assemblies. Whipping and vibration and their elimination are discussed. The author explains fully the use of the flexible fabric joint with the Hotchkiss drive. 304

(Discussed by Crawford, McComb, Meyers, Mott, Padgett, Reeser, Smith, Thoms and Vost)

SCHIPPER, J EDWARD—*Present Requirements of the Automobile User*

The consumption of metals by the war has reached such a stage that conservation of metal in commercial industries has become highly necessary. Fortunately, the tendency in automobile design has also been in the nature of metal conservation as a means of reducing weight, thereby securing economy of materials.

The public is demanding an efficient car in which every inch of wheelbase will be utilized and return a high percentage of riding comfort and space, and every unit of horsepower will be utilized

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for propulsion. The trend is toward a smaller car of less weight of car per pound of passenger weight and higher output per pound in every direction, whether it be engine, transmission system or body. 266

(Discussed by Strickland and Williams)

SCHIPPER, J EDWARD—*Recent Trends in Automobile Design*

War has turned an X-ray on the automobile industry with the result that only as a unit of efficient transportation can the automobile be considered as a wartime commodity. According to the author, the records of the automobile show that as an efficient transportation medium it falls short of wartime requirements. In spite of the great progress we have made, we have as yet no real wartime cars.

Modern tendencies have happily been along the proper lines and show increased use of overhead valves, shorter wheelbase with the same amount of body space, detachable cylinder heads, better cold-weather starting, clutches with fewer adjustments, and the increased use of forged parts.

The paper is illustrated by charts showing graphically the trend of design over the period of the last eight years. 277

STRATFORD, C W—*Standard Specifications for Lubricating Oils*

This paper is a result of the aid given by the Bureau of Standards, Bureau of Mines, the Army, the Navy and the Council of National Defense, and of the author's knowledge of internal-combustion phenomena and his experience in the art of refining oils.

For years engineers have been hoping for some measure of an oil's value as a lubricant in a particular service. The great war has given an opportunity to study the problem under the auspices of the Government, which has made it possible to develop this contribution. 511

STRITE, G T—*Fundamentals of Tractor Design*

To any one who has attended the annual demonstrations for the last five years the movement toward one general type of tractor is very apparent. While there will of course always be special machines for special uses, the general trend is unmistakably toward two types, the wheel and the tracklaying. The wheel tractor has come back to almost the identical design of the first tractor really made and sold. It is however smaller in size and refined in detail, developed into a satisfactory farm and manufacturing utility.

In this general trend are included such fairly constant factors as higher speed and lighter weight. In this connection the author introduces two tables, one showing the several factors affecting tractor design; the other, the characteristics of ten well-known makes of tractors. Already in some few makes the practical limit between weight and drawbar pull in pounds seems to have been reached. It takes practically the same power to pull two plows at three miles per hour as it does to pull three plows at two miles per hour, but there is a great difference in the traction required, the latter demanding 33 per cent more traction, either more weight or more surface on the ground. Long, sharp effective lugs increase traction, but they also absorb power and increase the relative difference between drawbar horsepower and the engine. In the future, there must be a practical limit set on horsepower in relation to weight.

From now on the work on the tractor will be a matter of refinement and development. The future standard tractor is not work for the inventor, but requires good sound practical engineering.

(Discussed by Dasey, Davis, Goldberger, Horning, Kettering and Monahan) 196

STURGESS, JOHN—*The Case for the Steam Car*

The title of this paper clearly conveys the character of the author's argument. At the outset he frankly recognizes the thought uppermost in the mind of the average engineer, "Why should steam cars be built?"

Direct answers are given to this question, the first being that full cognizance of the aims and laudable progress of gas-car manufacturers has been one of the strong incentives to the con-

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tinued construction of steam cars, based on the conviction of steam car advocates that the goal toward which gas-car builders are obviously striving can never be attained with internal-explosion engines.

The article proceeds with a technical comparison of the fundamental engineering features of both steam and gas cars, and defines the term "reserve power." The features that appeal directly to the motorist while operating his car are dealt with in detail, being followed by those considerations that concern the motorist in his capacity as owner.

The rest of the article covers the constructional and operating features of engine, boiler and burner, outlines the steps that have been taken leading to the adoption of the present types, and closes with an earnest warning to the industry to give careful thought to the author's contention "that steam is not only the shortest, but indeed, the only route to automobile perfection." 357

(Discussed by Brown, Castle, Emory, Gibson, Schlesinger, Thayer, Tillson and Warren)

TRASK, CHARLES A—*Tractor Friction Transmissions*

Under the headings, power-transmitting capacity, application of pressure, slippage, operating speeds, design and installation, the author has covered the problems in friction transmissions, illustrating them with charts and formulas. The difficulties in practical application and some misconceptions on the part of engineers are brought out and discussed.

Under friction transmission types the application of the so-called friction type to a number of tractors and industrial locomotives is explained and well illustrated. 320

(Discussed by Briggs, Crawford, Gallup, Hamerstadt and Weidely)

WARREN, PRESCOTT—*Pioneer Automobile Development*

The author holds that steam is the logical power for automobile propulsion, since it is the only power that meets the variable-speed variable-load requirements of wheeled traffic naturally. Not only is the performance of the steam car superior, but the constructional and operative means by which it is attained, are vastly simpler.

The first automobiles were steam cars, burning hard fuel. Liquid fuel, discovered half a century ago, made possible the getting of "power direct from fuel" by exploding it. The plan seemed feasible and caught the engineering fancy of the day. The cycle then seemed simpler than the steam cycle, but is, in fact, more complex, when delivery of power to the wheels is considered.

Following the lead of the engineers, the public came to accept the internal-explosion engine as the correct and final type. The author contends that this assumption on the part of makers and public alike, arises from the fact that both have, broadly speaking, totally ignored the steam car, which yields the very performance that the public is demanding and the makers are trying hard to give—that is, simplicity of construction; simplicity of functions, simplicity of control without gearshift or clutch; and with instantaneous response to the driver's will. Above all, it gives a sense of security and mastery. This performance, the author contends, can be had only with steam. 379

VINCENT, MAJOR J G—*Reasons Behind the Liberty Aviation Engine*

Last spring the Government had before it the problem of deciding on an aircraft program. There was no engine in this country developing enough power to satisfy requirements at the front. Abroad, the manufacturers did not have a type with which they were satisfied. It was decided to design a completely new engine and have this manufactured in quantity from Government drawings and specifications. On June 4th the author and Major Hall were ordered to produce ten sample engines. The first was ready for inspection in twenty-one days.

Around this brief statement the author tells the story of the now historic Liberty engine. Operating possibilities and the mechanical details of construction are covered to some extent also by the paper, although the reasons influencing the engineering choice and the pro and con of arguments have chief attention. The exceptional and vivid interest of both engineers and manufacturers in this creation for such it must be called—is clearly brought out.

(Discussed by Clark and Crane) 96

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ZIMMERMAN, J G—*Magneto Ignition for Farm Tractors*

The main object of the paper is to show several important changes necessary in tractor design to secure the best service conditions as regards the magneto. The paper emphasizes the need of great care and skill in installing the ignition equipment so that best advantage may be taken of the electric current generated to secure both accuracy of firing and uniformity of sparks in each cylinder.

The features brought out comprise in substance the following:

Accurate firing is essential to best power output.

Wiring must be uniform in character so that there will be no lag of spark due to different capacities of different cables.

Metallic conductor tubes weaken the "kick" or breakdown voltage at the spark-plug. Cables bound too closely together have the same effect.

The spark requirements for good ignition are a snappy kick voltage and a rapid follow-up arc, together with accurate spacing for the firing.

The best energy in the spark is obtained when the plug gap corresponds to maximum watts developed by magnetos. This is roughly the gap that will require, at the compression used, a kick voltage equal to one-half the kick voltage developed by the magneto on open circuit at the speed considered.

The make-and-break spark usually has more energy than the jump spark obtained from magnetos of the same dimensions. This type of spark has a greater follow-up intensity also, resulting in a greater total igniting power for differently ignited fields. The disadvantage lies mainly in the relative bulkiness of parts required and the necessity for moving parts through cylinder walls. The main difficulty as to spark value is that the spark is not of the same snappy character.

The position of a magneto is a matter of great importance, since it needs attention occasionally and should be protected from heat and dirt. 387

(Discussed by Hatch, Oetting and Smith)

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ANNUAL MEETING

OF THE

SOCIETY OF AUTOMOTIVE ENGINEERS

NEW YORK CITY

January 9, 1918

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MEETING OF THE STANDARDS COMMITTEE

The Standards Committee of the Society met Jan. 9, 1918, in New York, with Chairman John G. Utz presiding, to pass on the work accomplished since the June meeting. Some sixty-five different recommendations were submitted for acceptance by the Committee. The majority of these were submitted by the Aeronautic Division, although reports were submitted also by the Motorcycle, Roller-Chain, Tractor, Marine, Data Sheet, Lighting, Tire and Rim, and Miscellaneous Divisions.

The reports as finally adopted are given on the following pages.

In the discussion of the Aeronautic Division report at the Committee meeting a number of minor changes were suggested; these have been incorporated in the report as finally approved. It was decided to refer back the glue specifications† to the Division, inasmuch as the test for double shear had been found to be not acceptable, and moreover there seemed to be considerable uncertainty as to the actual strength of the glue and of the wood to be used in the test for adhesiveness.

The report of the Motorcycle Division covered matters especially applicable to military motorcycles. The work was first undertaken by an informal committee under the auspices of the Society, and was then continued by the formation of a Motorcycle Division of the Standards Committee. After the report presented by the Division was prepared the Government designed a standardized motorcycle, using the work of the Division in the design.

The Roller-Chain Division has been working with a committee from the American Society of Mechanical Engineers, and the recommendations submitted on nomenclature and dimensions represent the opinions of both the Division and the A. S. M. E. Committee. The roller-chain dimensions are not new sizes, but are simply introduced to furnish sizes most suitable for general work. It was said there are about five times as many sizes of the roller chains in general use as are actually required. A start has been made by designating the pitch, chain width, and roller diameter; it is expected that detail di-

†See Report of Glue Committee, Vol. I. No. 4. THE JOURNAL

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mensions to correspond can be added later if conditions in the industry warrant such a step. The Division will continue work on tolerances for the chain dimensions, and also proposes to work out with the cooperation of foreign roller-chain manufacturers a standardized series of metric chains.

The Tractor Division, in addition to submitting a recommendation that the tractor drawbar connection should be mounted in a horizontal position, recommended that a number of present S. A. E. standards be used for tractor work.

The Data Sheet Division submitted a report showing progress.

A general index, carefully cross-indexed, was prepared for the data sheets in the S. A. E. Handbook, and was mailed to all the members of the Society, together with the latest data sheets.

The Division recommended that separate index sheets or sections of the general index include the general standards and recommended practices that may be approved for use in the following subjects:

Aeronautic, Marine, Motorcycle, Stationary and Farm Engine, and Tractor.

The Miscellaneous Division had previously suggested that the dimensions for adjustable yoke rod-ends, plain yoke rod-ends, and eye rod-ends, be extended to include ⅞ and 1-in. sizes. A standard set of dimensions for mounting bumpers was also adopted, as was a complete set of screw-thread pitches for all diameters from ¼ up to 6 in. and over. Regular and fine pitches were specified for each size of bolt, with the result that a systematic standard of screw threads is now offered.

The report of the Marine Division includes dimensions for the flanges for coupling the reverse gear to the engine, for coupling the propeller shaft to the reverse gear, for use with the extended shaft, and for the propeller mounting. These apply to sizes of propellers up to 4-in. shaft diameter.

On account of the lateness of the hour it was necessary to adjourn the meeting of the Standards Committee before taking action on the reports presented by the Marine, Tire and Rim, and the Lighting Divisions. The dimensions proposed by the Marine Division and the report of the Tire and Rim Division were accepted at the special meeting of the Standards Committee called during the Society meeting Thursday morning. The Lighting Division report was accepted by the Standards Committee with some alterations.

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THIRD REPORT OF AERONAUTIC DIVISION

As adopted by the Society

The report of the Aeronautic Division includes specifications for cellulose acetate dopes, cellulose nitrate dopes, spar varnish, round high-strength steel wire, non-flexible 19 steel-wire cable, flexible 6 by 7 steel-wire cable, flexible 7 by 7 steel-wire cable, extra flexible, 7 by 19 steel-wire cable, reels for cable, tachometer drive, rubber hose for gasoline, and plain hexagon nuts (for bodies and wings). The dimensions approved for ball hex, ball castle nuts, castle hexagon nuts, ball hexagon-head bolts and plain hexagon-head bolts at the June meeting were extended to include the 9/16 and 5/8-in. sizes. The clamps and fittings for 3/16 to 3-in. rubber hose* already standardized were approved for aeronautic practice. Complete specifications were recommended for airplane engine testing forms and for engine-weight specifications. Other matters adopted are spark-plug shell dimensions and bevel-washer dimensions.

Dopes—Cellulose Acetate

The temperature of the doping rooms of airplane factories should be maintained at 70 deg. fahr. The airplane manufacturers should store the dope under a roof.

(1) *Definition* Dopes for airplane fabrics must consist of a clear, uniform mixture of ingredients and be capable of shrinking the fabric to the degree of tautness desired by the purchaser's inspection. The residual film should be reasonably transparent, and free from white spots, and should give a smooth homogeneous surface, when applied in a horizontal position in an atmosphere not exceeding 65 per cent humidity and 75 deg. fahr. temperature, free from direct draft.

(2) *Viscosity* The viscosity of the dope must permit of direct application without dilution at a temperature not lower than 60 deg. fahr.

(3) *Coating* The dope, when dry, must adhere to the fabric with sufficient tenacity to prevent peeling off in sheets. Test strips should show lint attached to the side that has been in contact with the fabric.

(4) *Effect on Tensile Strength and Weight* Four coats or an equivalent of the dope, 48 hr. after application, must increase the tensile strength of linen fabrics

*See report of Miscellaneous Division, Part I, 1916 S. A. E. Transactions or Data Sheet 37, Vol. I, S. A. E. Handbook

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not less than 25 per cent of the original average strength of warp and filling, and of cotton fabric not less than 15 per cent. The increase in weight per square yard of doped fabric should not be less than 2 nor greater than 2.75 oz. The test should be made under standard conditions of humidity and temperature on standard fabrics.

(5) *Acidity* No mineral acids may be present in the dope, and the amount of free organic acid figured as acetic acid may not exceed 0.2 per cent. No compounds may be present that would be injurious to the fabric.

(6) *Sulphates* Dopes that show the presence of free sulphuric acid by the test given below are not acceptable.

(7) *Tetrachlorethane* Dopes containing tetrachlorethane will not be acceptable for repair work, but will be permissible in factories provided with adequate ventilation.

(8) *Cellulose Acetate* The cellulose acetates used should contain no free mineral acid and not more than 0.1 per cent free acetic acid, and should be stable. The amount of cellulose acetate shall not be less than 60 grams per liter of dope.

(9) *Solvents* The volatile solvents employed should present no danger to the workman applying them.

(10) *Inflammability* Five drops of gasoline dropped on the film that has been dried for 48 hr. and immediately ignited should have no more serious effect than to char the fabric under the moistened section of the film.

(11) *Exposure Test* Dopes must comply with the following test: A square frame, 12 by 12 in. inside measurement, is covered on both sides with fabric, the fabric being tacked to the outer side of the frame. The fabric is to be tacked under uniform tension, simulating that employed in airplane manufacture. Four coats or an equivalent of dope are to be applied to each side of the frame, each coat being allowed to dry thoroughly before the succeeding coat is applied. The frames are to be exposed on a roof in an unshaded horizontal position, one side being constantly uppermost. After 60 days of constant exposure no spontaneous cracking of the doped surface should be apparent, and after remaining one hour at a temperature of from 70 to 80 deg. fahr. the film shall not crack and shall have a decided ring. This test shall be made in duplicate and comparatively with a dope that has previously passed the test, and shall be in effect until a mechanical test is adopted.

(12) *Shipment* Dope shall be shipped in metal cans,

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metal or wooden barrels, or earthenware containers. Inspection of the containers shall be permitted to insure against the accidental introduction of foreign material. The container shall be marked with the date of manufacture, serial number, gross tare and net weight.

Testing of Dopes

(1) *Acidity* A 500-cc. beaker, containing about 200 cc. of water is counterbalanced on a large balance. The balance is adjusted to one hundredth gram by adding or removing water. About 10 grams of dope are poured into the water and the increase in weight noted. This is rapidly done to 0.01 gram to diminish solvent loss. The dope is stirred up and allowed to stand 10 to 15 min., with occasional stirring. The liquid is decanted through a rather porous filter into an 800-cc. beaker and 150 cc. of warm water added to the residue. It is allowed to stand 10 to 15 min. with frequent stirring, and poured through the filter into the 800-cc. beaker. The residue is washed with 150 cc. of warm water as before. A few drops of phenolphthalein are added and the solution titrated with twenty-fifth normal caustic soda to a color that persists for one-half minute.

Some dopes, notably those containing much acetone, when poured into water precipitate as a milky solution containing shreds of the acetate. The resulting liquor filters slowly and passes through the filter paper in a cloudy condition. Since the acetate is finely divided, it is practically free from acetic acid, and additional washing is unnecessary. The end-point is not quite as sharp as when all the acetate has been removed, owing to hydrolysis of the suspended material, but is sufficiently accurate for all practical purposes. Absence of mineral acids must be proved by qualitative tests.

(2) *Sulphates* Twenty grams of cellulose acetate dope are treated with 150 cc. of water in a pressure bottle at 100 deg. cent. for 24 hr. The resulting liquor is filtered and tested for free sulphuric acid.

(3) *Amount of Cellulose Acetate* Pour 25 grams of the dope into a Petri dish 6 in. in diameter, and evaporate to dryness on the steam bath. Extract the residue with ether in a Soxhlet until all extractive material has been removed. Dry at 60 deg. cent. to constant weight, and weigh.

(4) *Film* Pour some of the dope on a glass plate and allow to dry spontaneously. The film may be examined for the general characteristics of transparency, coherence, strength and flexibility.

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Dopes—Cellulose Nitrate

(1) *Definition* Dopes for airplane fabrics must consist of a clear, uniform mixture of ingredients, and be capable of shrinking the fabric to the degree of tautness desired by the purchaser's inspection. The residual film should be reasonably transparent and free from white spots, and should give a smooth, homogeneous surface when applied in a horizontal position in an atmosphere not exceeding 65 per cent of humidity and 75 deg. fahr. temperature, free from direct draft.

(2) *Viscosity* The viscosity of the dope must permit of direct application without dilution at a temperature not lower than 60 deg. fahr.

(3) *Coating* The dope, when dry, must adhere to the fabric with sufficient tenacity to prevent peeling off in sheets. Test strips should show lint attached to the side that has been in contact with the fabric.

(4) *Effect on Tensile Strength and Weight* Four coats or an equivalent of the dope, 48 hr. after application must increase the tensile strength of linen fabrics not less than 25 per cent of the original average strength of warp and filling and of cotton fabrics not less than 15 per cent. The increase in weight per square yard of doped fabric should not be less than 2 nor greater than 2.75 oz. The test shall be made under standard conditions of humidity and temperature on standard fabrics.

(5) *Acidity* No mineral acids shall be present in the dope and the amount of free organic acidity figured as acetic acid may not exceed 0.05 per cent. No compounds may be present that would be injurious to the fabric.

(6) *Cellulose Nitrate* The cellulose nitrate used in the manufacture of dope shall be purified and give a negative potassium iodide test at the end of 20 min., according to the standard Government method for the Abel stability test. The amount of cellulose nitrate used shall be not less than 55 grams per liter of dope.

(7) *Solvents* The volatile solvents employed shall present no danger to the workman applying them.

(8) *Exposure Test* Dopes must comply with the following test: A square frame, 12 by 12 in. inside measurement, is covered on both sides with fabric, the fabric being tacked to the outer side of the frame. The fabric is to be tacked under uniform tension, simulating that employed in airplane manufacture. Four coats or an equivalent of dope are to be applied to each side of the frame, each coat being allowed to dry thoroughly before

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the succeeding coat is applied. The frames are to be exposed on a roof in an unshaded horizontal position, one side being constantly uppermost. After 60 days of constant exposure no spontaneous cracking of the doped surface should be apparent, and after remaining one hour at a temperature of 70 to 80 deg. fahr., the film shall not crack and shall have a decided ring. This test shall be made in duplicate and comparatively with a dope that has previously passed the test, and shall be in effect until a mechanical test is adopted.

(9) *Shipment* Dope shall be shipped in metal cans, metal or wooden barrels or earthenware containers. Inspection of the containers shall be permitted to insure against the accidental introduction of foreign materials. The container shall be marked with the date of manufacture, serial number, gross tare and net weight.

Testing of Dopes

(1) *Acidity* A 500-cc. beaker, containing about 200 cc. of water, is counterbalanced on a large balance. The balance is adjusted to one-hundredth gram by adding or removing water. About 10 grams of dope are poured into the water and the increase in weight noted. This is rapidly done to 0.01 gram to diminish solvent loss. The dope is stirred up and allowed to stand 10 to 15 minutes with occasional stirring. The liquid is decanted through a rather porous filter into an 800-cc. beaker, and 150 cc. of warm water added to the residue. It is allowed to stand 10 to 15 minutes with frequent stirring, and poured through the filter into the 800-cc. beaker. The residue is washed with 150 cc. of warm water as before. A few drops of phenolphthalein are added and the solution titrated with twenty-fifth-normal caustic soda to a color that persists for one-half minute. Any satisfactory substitute method will be permissible on approval.

(2) *Amount of Cellulose Nitrate* Pour 25 grams of the dope into 100 grams of chloroform, stirring constantly. Extract in a soxhlet with chloroform until all material possible has been removed. Dry at 60 deg. cent. to constant weight, and weigh.

(3) *Film* Pour some of the dope on a glass plate and allow to dry spontaneously. The film may be examined for the general characteristics of transparency, coherence, strength, and flexibility.

Spar Varnish Specification

Composition and General Properties The material shall be the best long oil varnish, suitable for application

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on wood, "doped" linen or cotton, and metal, and resistant to air, light and water. The manufacturer is given the greatest latitude in the selection of raw materials and process of manufacture in order to produce a product of the highest quality.

Physical Characteristics The material shall comply with the following requirements:

(1) It shall be clear and transparent.

(2) Its color shall be no darker than a standard color solution, made by dissolving 6 grams of pure powdered potassium bichromate in 100 cc. of pure concentrated sulphuric acid (specific gravity 1.84). Gentle heat may be used, if necessary, to secure a perfect solution of the bichromate. The color comparison will be made by placing the varnish and the standard color solution in clear, thin-walled glass tubes of the same diameter, 1.5 to 2 cm. (5/8 to 13/16 in.) to a depth of at least 2.5 cm. (1 in.) and comparing the colors by looking through the tubes, across the column of the liquid by transmitted light.

(3) The flash point shall not be below 35 deg. cent. (95 deg. fahr.) when determined in an open tester.

(4) The varnish will be flowed on one side of a 10 by 15-cm. (approx. 4 by 6 in.) panel of bright tin. The panels* shall be approximately 0.3 to 0.4 mm. (0.0125 to 0.0158 in.) thick, and shall be cleaned thoroughly with benzol. When a panel is held in a vertical position and maintained at a temperature of 21 to 32 deg. cent. (70 to 90 deg. fahr.), the varnish shall set to touch at a point not less than 2.5 cm. (1 in.) from the side or top edges of the film, in not more than 5 hr.; and shall dry hard in not more than 24 hr. to a clear, hard, glossy film. The panel will then be allowed to dry for a further period of not less than 5 days and then brought to a temperature of not less than 21 deg. cent. (70 deg. fahr.) nor more than 24 deg. cent. (75 deg. fahr.) and maintained at this temperature for not less than 15 min. The panel, with the varnished side on the outside, will then be bent double rapidly over a rod 3 mm. (1/8 in.) diameter. The varnish film shall show no cracking or flaking at the point of bending.

(5) The varnish shall be applied to a basswood panel which has been filled with one coat of drop black in oil thinned with turpentine and drier, and allowed to dry for not less than 10 days before applying the varnish. It shall have suitable body to give proper brushing, flowing

*The base metal must weigh from 90 to 100 lb. per standard box of 112 sheets, 14 by 20 in. No. 30 to 28 U. S. standard plate gage

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and covering properties. The first coat of varnish will be allowed to dry 48 hr., then be lightly sandpapered, a second coat applied and allowed to dry 72 hr. The panel will then be inclined at an angle of about 45 deg. and a gentle stream of cold tap water allowed to flow down the middle of the panel for 18 hr. After wiping off with a chamois skin any deposits due to the tap water, the varnish shall show no whitening, dulling or other defects. A small stream of boiling distilled water shall then be allowed to flow down another portion of the panel for 20 minutes. The water will be siphoned onto the surface of the panel through a small glass tube directly from a container in which it is boiling, in such a manner that there will be no appreciable lowering of the temperature of the water before it touches the varnish film. The siphon delivery tube will be in a plane nearly parallel to the plane of the panel, so that the impact of the water will not tend to break the film. The varnish shall show no appreciable whitening and no more than a very slight dulling or other indications of marked deterioration, either when observed immediately after removal from the water, or after drying for 2 hours.

(6) The varnish shall be applied in three coats to two unfilled panels of maple wood, not less than 14 by 45 by 2 cm. (5½ by 18 by ¾ in.), allowing three days for the drying of each coat. The first coat, after drying indoors for 3 days will be sandpapered lightly with No. 00 sandpaper before the application of the next coat. The second and third coats will not be sandpapered or rubbed, and the duplicate panels will be exposed outdoors, 45 deg. to the vertical, facing south, three days after the application of the finishing coat. The backs and edges of the panels will also be varnished with three coats of the same sample, but for these surfaces the details of the method of application as given need not be adhered to, and the effects of exposure on these surfaces will not be considered.

On this test, the varnish shall show satisfactory durability and weather resistant properties. In cases where the award of a contract cannot be delayed for the results of the exposure test, award may be made on the basis of the other requirements; but a varnish of any specific brand that does not show up satisfactorily on exposure test may be omitted from consideration in future awards, and a preliminary submittal of samples for making exposure tests may be called for.

NOTE—In general, results comparable with those given by the above method may be obtained by flowing one

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coat of varnish on duplicate panels of bright tin thoroughly cleaned with benzol, allowing it to dry 48 hr., and then immersing one of the panels in distilled water at room temperature for 18 hr., and the other panel in boiling distilled water for 15 minutes. But the test on wood should be made in all cases when there is any doubt regarding the resistance of the varnish to water.

Round High Strength Steel Wire

This specification covers solid high-strength steel wire round section, used in the construction of aircraft and in which flexibility is of minor importance.

Workmanship and Finish The wire shall be cylindrical and smooth and may show no evidence of scrapes, splints, cold shuts, rough tinning, or other defects not in accordance with best commercial practice.

Tensile Test Samples for the tensile test shall be not less than 15 in. long and free from bends and kinks. In making tensile tests on aircraft wire, the distance between jaws of testing machine, with the sample in place and before test, shall be 10 in. The wire must not break at less than the amount specified in the accompanying table, which is a part of this specification.

Torsion Test Samples for the torsion test shall be straight and not less than 10 in. long. The sample shall be gripped by two vises 8 in. apart; one vise shall be turned uniformly at a speed not exceeding 60 r.p.m. (on the larger sizes of wire this speed shall be reduced sufficiently to avoid undue heating of the wire). One vise shall have free axial movement in either direction. All wire shall be required to withstand a number of complete turns, which are to be agreed upon between the purchaser and the manufacturer.

Bend Test Samples for bend test shall be straight and not less than 10 in. long. One end of the sample shall be clamped between jaws having their upper edges rounded with 3/16 (0.188) in. radius. The free end of the wire shall be held loosely between two guides and bent 90 deg. over one jaw; this is to be counted as one bend. On raising to a vertical position the count will be two bends. The wire shall then be bent to the other side, and so forth, alternating to fracture. The minimum number of bends required is stated in the table.

Wrapping Test This is to be made on at least 10 per cent of the total number of coils offered for inspection at one time. The wire is wrapped around its own diameter eight consecutive turns with a pitch substantially equal to the diameter of the wire and then un-

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wrapped, maintaining the free end at approximately 90 deg. with the mandrel. It must stand this test without fracture. Because of the possibility of personal error in making this test, failure on one test is not considered conclusive, and if required to do so the inspector shall make at least one, but no more than two, additional tests on the sample of wire. If any of these tests is successful, the material shall be passed as satisfactory in this respect.

DATA FOR ROUND HIGH STRENGTH STEEL WIRE

American Wire Gage No.	Diameter, In.	Weight, Lb./100 Ft.	Number of Bends Through 90 Deg.	Minimum Breaking Strength, Lb.	Tensile Strength, Lb./Sq. In.
6	0.162	7.010	5	4,500	219,000
7	0.144	5.560	6	3,700	229,000
8	0.129	4.400	8	3,000	233,000
9	0.114	3.500	9	2,500	244,000
10	0.102	2.770	11	2,000	244,000
11	0.091	2.200	14	1,620	254,000
12	0.081	1.744	17	1,300	252,000
13	0.072	1.383	21	1,040	255,000
14	0.064	1.097	25	830	258,000
15	0.057	0.870	29	660	259,000
16	0.051	0.690	34	540	264,000
17	0.045	0.547	42	425	267,000
18	0.040	0.434	52	340	270,000
19	0.036	0.344	70	280	275,000
20	0.032	0.273	85	225	280,000
21	0.028	0.216	105	175	284,000

Selection of Test Specimen A tensile, a torsion and a bend test shall be made on each end of each piece or coil of wire. When an individual coil of wire is to be divided into smaller coils to meet special requirements, it is sufficient to make one test on the original coil and to cut and seal the small coils in the presence of the inspector.

Dimensions and Tolerances All wire for this purpose shall be furnished in decimal sizes corresponding to the American wire gage (Brown & Sharpe gage).

A permissible variation of 0.002 in. above gage on all sizes will be accepted, but no wire will be accepted having a variation of more than 0.0005 in. below gage.

NON-FLEXIBLE 19 STEEL WIRE CABLE

This specification covers high-strength 19 steel wire cable used in the construction of aircraft and in which flexibility is of minor importance.

Manufacture The steel wires composing the cable shall be laid around the center wire in one or two layers as required by the number of wires in the cable with a left-hand (counter-clockwise) pitch and with a length of

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lay not to exceed eleven times the diameter of the cable or not less than nine times the diameter of the cable.

Wires composing the cable shall be uniformly coated with pure tin or galvanized* to solder readily.

The joints in wires composing the cable shall be brazed in a gas fire. Tucked in, welded or twisted joints will not be permitted. No two brazed joints in individual wires shall be closer to one another than 150 ft. in the completed cable. All brazed joints in wires shall be tinned. Exposed brass at joints shall not constitute cause for rejection.

Workmanship and Finish Each length of cable is to be evenly laid and free from kinks, loose wires or other irregularities. The cable shall remain in this condition when unwound from the reel or bent around a standard thimble, proper precautions being taken to secure the ends.

Tensile Test A tensile test shall be made upon each reel of cable purchased of a size.

Samples of cable for testing for tensile strength shall be not less than 24 in. long. In making tensile tests the distance between jaws of testing machine, with sample in place and before test, shall be not less than 10 in.

Samples for tensile test may be clamped in the jaws of the testing machine in the usual manner to facilitate testing, but in case of failure or dispute on individual tests, and at the request of the manufacturer, check tests shall be made by socketing the samples with pure zinc.

Cable for use in the construction of aircraft shall meet the required breaking strength specified in the table.

Bend Test One bend test is to be made on a sample cut from each reel of cable of a given size. Each sample must be bent once around its own diameter and straightened again at least 20 times in succession in the same direction of bending without any of the wires breaking.

Torsion Test This is to be made on one wire from each sample of cable taken for tensile test. The wire is to be gripped by two vises 8 in. apart. One vise shall be turned uniformly at as high a rate of speed as possible without perceptibly heating the wire. One vise shall have free axial movement in either direction.

The number of complete turns which the wire shall stand is determined by the relation:

Number of turns = 2.2 ÷ dia. in inches.

Failure of one piece of wire to show the full number of

*If the galvanized coating is used, the method is to be specified by the purchaser

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turns specified in the above torsion test shall not be considered cause for rejection, but in such case two additional tests shall be made on two more wires from the same sample of cable, and if both samples meet the requirements of the specifications the cable shall be accepted in this respect.

Dimensions and Tolerances There shall be no permissible variation in gage below size. Cable having a diameter of 0.031 (1/32) to 0.156 (5/32) in., inclusive, shall have a permissible variation of 10 per cent above size, and cable having a diameter of 0.188 (3/16) to 0.375 (3/8) in., inclusive, shall have a permissible variation of 7 per cent above size.

WEIGHTS, SIZES AND STRENGTH OF NO. 19 CABLE

Diameter, In.	Minimum Breaking Strength, Lb.	Approx. Weight, Lb./100 Ft.
0.312 (5/16)	12,500	20.65
0.250 (1/4)	8,000	13.50
0.218 (7/32)	6,100	10.00
0.188 (3/16)	4,600	7.70
0.156 (5/32)	3,200	5.50
0.125 (1/8)	2,100	3.50
0.109 (7/64)	1,600	2.60
0.094 (3/32)	1,100	1.75
0.078 (5/64)	780	1.21
0.062 (1/16)	500	0.78
*0.031 (1/32)	185	0.30

*No. 7 wire.

FLEXIBLE 6 BY 7 STEEL WIRE CABLE

This specification covers high strength 6 by 7 flexible steel wire cable used in the construction of aircraft and in which flexibility is important.

Manufacture The steel wires composing the individual strands of the cable shall be laid concentrically around the center wire in one layer of six wires with a left-hand (counter-clockwise) pitch or lay. The cable itself shall be constructed by twisting six of these strands composed of seven wires each around a cotton center with a right-hand (clockwise) pitch or lay of six to eight times the diameter of the whole.

Wires composing the cable shall be uniformly coated with pure tin to solder readily.

Joints in wires in cable having a diameter of 0.156 (5/32) in. and larger shall be brazed in a gas fire. In cable having a diameter of 0.125 (1/8) in. or less, wires may be joined either by brazing or twisting, at the manufacturer's convenience. Tucked-in or welded joints are not permitted. No two joints in individual wires shall

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be closer than 30 ft. to one another in the completed cable. All brazed joints shall be tinned. Exposed brass at joints shall not constitute cause for rejection.

Workmanship and Finish Each length of cable is to be evenly laid, and free from kinks, loose wires or other irregularities. The cable shall remain in this condition when unwound from the reel or bent around a standard thimble, proper precautions being taken to secure the ends.

Tensile Test A tensile test shall be made upon each individual reel of cable purchased of a size.

Samples of cable for testing for tensile strength shall be not less than 24 in. long. In making tests the distance between jaws of testing machine with sample in place and before test shall be not less than 10 in.

Samples for tensile test may be clamped in the jaws of the testing machine in the usual manner to facilitate testing; but in case of failure or dispute on individual tests and at the request of the manufacturer check tests shall be made by socketing the samples with pure zinc.

Cable for use in the construction of aircraft shall meet the required breaking strength specified in the table.

Bend Test One bend test is to be made on a sample cut from each reel of cable of a given size. Each sample must be bent once around its own diameter and straightened again at least 20 times in succession in the same direction of bending without any of the wires breaking.

Torsion Test This is to be made on one wire from each sample of cable for tensile test. The wire is to be gripped by two vises 8 in. apart. One vise shall be turned uniformly at as high a rate of speed as possible without perceptibly heating the wire. One vise shall have free axial movement in either direction.

The number of complete turns which the wire shall stand is determined by the relation:

Number of turns $= 2.2 \div$ dia. in inches.

Failure of one piece of wire to show full number of turns specified in this torsion test shall not be considered cause for rejection, but in such case two additional tests shall be made on two more wires from the same sample of cable, and if both samples meet the requirements of the specifications the cable shall be accepted in this respect.

Dimensions and Tolerances There shall be no permissible variation in gage below size. Cable having a diameter of 1/16 to 3/32 in., inclusive, shall have a per-

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missible variation of 12 per cent above size; cable having a diameter of 1/8 to 3/16 in., inclusive, shall have a permissible variation of 10 per cent above size, and cable having a diameter of 7/32 to 3/8 in., inclusive, shall have a permissible variation of 7 per cent above size.

WEIGHTS, SIZES AND STRENGTH OF 6 BY 7 CABLE

Diameter, In.	Minimum Breaking Strength, Lb.	Approx. Weight, Lb./100 Ft.
0.312 (5/16)	7,900	15.00
0.250 (1/4)	5,000	9.50
0.218 (7/32)	4,000	7.43
0.188 (3/16)	2,750	5.30
0.156 (5/32)	2,200	4.20
0.125 (1/8)	1,150	2.20
0.109 (3/32)	830	1.50
0.094 (1/32)	780	1.30
0.078 (5/64)	480	0.83
0.062 (1/16)	400	0.73

FLEXIBLE 7 BY 7 STEEL WIRE CABLE

This specification covers high strength 7 by 7 flexible steel wire cable used in the construction of aircraft and in which flexibility is important.

Manufacture The steel wires composing the individual strands of the cable shall be laid concentrically around the center wire in one layer of six wires with a left-hand (counter-clockwise) pitch or lay. The cable itself shall be constructed by twisting six of these strands composed of seven wires each around a seventh strand of the same construction and material with a right-hand (clockwise) pitch and with a length of lay of six to eight times the diameter of the whole.

Wires composing the cable shall be uniformly coated with pure tin to solder readily.

Joints in wires in cable having a diameter of 0.156 (5/32) in. and larger shall be brazed in a gas fire. In cable having a diameter of 0.125 (1/8) in. or less, wires may be joined either by brazing or twisting, at the manufacturer's convenience. Tucked-in or welded joints are not permitted. No two joints in individual wires shall be closer to one another in the completed cable than 30 ft. All brazed joints shall be tinned. Exposed brass at joints shall not constitute cause for rejection.

Workmanship and Finish Each length of cable is to be evenly laid, and free from kinks, loose wires, or other irregularities. The cable shall remain in this condition when unwound from the reel or bent around a standard

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thimble, proper precautions being taken to secure the ends.

Tensile Test A tensile test shall be made upon each individual reel of cable purchased of a size.

Samples of cable for testing for tensile strength shall be not less than 24 in. long. In making tests the distance between jaws of testing machine with sample in place and before test shall be not less than 10 in.

Samples for tensile test may be clamped in the jaws of the testing machine in the usual manner to facilitate testing; but in case of failure or dispute on individual tests and at the request of the manufacturer, check tests shall be made by socketing the samples with pure zinc.

Cable for use in the construction of aircraft shall meet the required breaking strength specified in the table.

Bend Test One bend test is to be made on a sample cut from each reel of cable of a given size. Each sample must be bent once around its own diameter and straightened again at least 20 times in succession in the same direction of bending without any of the wires breaking.

Torsion Test This is to be made on one wire from each sample of cable for tensile test. The wire is to be gripped by two vises 8 in. apart. One vise shall be turned uniformly at as high a rate of speed as possible without perceptibly heating the wire. One vise shall have free axial movement in either direction.

The number of complete turns which the wire shall stand is determined by the relation:

Number of turns $= 2.2 \div$ dia. in inches.

Failure of one piece of wire to show full number of turns specified in the above torsion test shall not be considered cause for rejection, but in such case two additional tests shall be made on two more wires from the same sample of cable, and if both samples meet the requirements of the specifications the cable shall be accepted in this respect.

Dimensions and Tolerances There shall be no permissible variation in gage below size. Cable having a diameter of 1/16 to 3/32 in., inclusive, shall have a permissible variation of 12 per cent above size; cable having a diameter of 1/8 to 3/16 in., inclusive, shall have a permissible variation of 10 per cent above size, and cable having a diameter of 7/32 to 3/8 in., inclusive, shall have a permissible variation of 7 per cent above size.

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WEIGHTS, SIZES AND STRENGTH OF 7 BY 7 CABLE

Diameter, In.	Minimum Breaking Strength, Lb.	Approx. Weight, Lb./100 Ft.
0.312 (5/16)	9,200	16.70
0.250 (1/4)	5,800	10.50
0.218 (7/32)	4,600	8.30
0.188 (3/16)	3,200	5.80
0.156 (5/32)	2,600	4.67
0.125 (1/8)	1,350	2.45
0.094 (3/32)	920	1.45
0.078 (5/64)	550	0.93
0.062 (1/16)	480	0.81

EXTRA FLEXIBLE 7 BY 19 STEEL WIRE CABLE

This specification covers high strength 7 by 19 extra flexible steel wire used in the construction of aircraft and in which extra flexibility is important.

Manufacture The steel wires composing the individual strands of cable shall be laid concentrically around the center wire in one layer of six wires and another, or outer layer, of twelve wires with a left-hand (counterclockwise) pitch, the lay or pitch of both layers being of the same length; the cable itself shall be constructed by twisting six of these strands composed of nineteen wires each around a seventh strand of the same construction and material with a right-hand (clockwise) pitch or lay of six to eight times the diameter of the whole.

It is to be understood that the strand composing this cable must not necessarily be composed of wires all of the same diameter.

Wires composing the cable shall be uniformly coated with pure tin to solder readily.

Joints in wires in cable having a diameter of 0.188 (3/16) in. and larger shall be brazed in a gas fire. In cable having a diameter of 0.156 (5/32) in. or less, wires may be joined either by brazing or twisting at the manufacturer's convenience. Tucked-in or welded joints are not permitted. No two joints in individual wires shall be closer than 30 ft. to one another in the completed cable.

Workmanship and Finish Each length of cable is to be evenly laid and free from kinks, loose wires or other irregularities. The cable shall remain in this condition when unwound from the reel or bent around a standard thimble, proper precautions being taken to secure the ends.

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Tensile Test A tensile test shall be made upon each individual reel of cable purchased of a size.

Samples of cable for testing for tensile strength shall be not less than 24 in. long. In making tensile tests the distance between jaws of testing machine with sample in place and before test shall be not less than 10 in.

Samples for tensile test may be clamped in the jaws of the testing machine in the usual manner to facilitate testing, but in case of failure or dispute on individual tests, and at the request of the manufacturer, check tests shall be made by socketing the samples with pure zinc.

Cable for use in the construction of aircraft shall meet the required breaking strength specified in the table.

Bend Test One bend test is to be made on a sample cut from each reel of cable of a given size. Each sample must be bent once around its own diameter and straightened again at least 20 times in succession in the same direction of bending without any of the wires breaking.

Torsion Test This is to be made on one wire from each sample of cable taken for tensile test. The wire is to be gripped by two vises 8 in. apart; one vise shall be turned uniformly at as high a rate of speed as possible without perceptibly heating the wire. One vise shall have free axial movement in either direction.

The number of complete turns which the wire shall stand is determined by the formula:

Number of turns = 2.2 ÷ dia. in inches.

Failure of one piece of wire to show full number of turns specified in the above torsion test shall not be considered cause for rejection, but in such case two additional tests shall be made on two more wires from the same sample of cable, and if both samples meet the requirements of the specification the cable shall be accepted in this respect.

WEIGHTS, SIZES AND STRENGTH OF 7 BY 19 CABLE

Diameter, In.	Minimum Breaking Strength, Lb.	Approx. Weight, Lb./100 Ft.
0.375 (⅜)	14,400	26.45
0.344 (11/32)	12,500	22.53
0.312 (5/16)	9,800	17.71
0.281 (9/32)	8,000	14.56
0.250 (¼)	7,000	12.00
0.218 (7/32)	5,600	9.50
0.188 (3/16)	4,200	6.47
0.156 (5/32)	2,800	4.44
0.125 (⅛)	2,000	2.88

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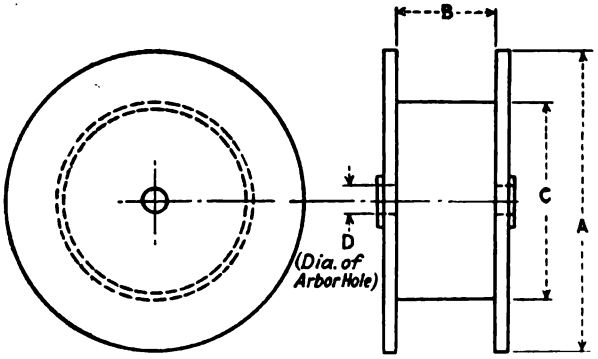
Dimensions and Tolerances There shall be no permissible variation in diameter below size. Cable having a diameter of 0.125 (1/8) to 0.187 (3/16) in., inclusive, shall have a permissible variation of 10 per cent above size, and cable having a diameter of 0.218 (7/32) to 0.375 (3/8) in., inclusive, shall have a permissible variation of 7 per cent above size.

Reels for Aircraft Cable



Diameter Cable, In.	A	B	C	D	A	B	C	D	A	B	C	D	A	B	C	D
	1,000 Feet				3,000 Feet				5,000 Feet				10,000 Feet			
1/32	12	4	8	1⅛	12	4	8	1⅛	12	4	8	1⅛	16	4	10	1⅛
1/16	12	4	8	1⅛	12	4	8	1⅛	16	4	10	1⅛	16	7	12	1⅛
5/64	12	4	8	1⅛	16	4	10	1⅛	16	7	12	1⅛	16	10	8	1⅛
3/32	12	4	8	1⅛	16	4	10	1⅛	16	7	12	1⅛	16	10	8	1⅛
7/64	16	4	10	1⅛	16	7	12	1⅛	16	10	8	1⅛	18	10	8	1⅛
1/8	16	4	10	1⅛	16	7	12	1⅛	16	10	8	1⅛	24	10	10	1⅛
9/64	16	7	12	1⅛	16	10	8	1⅛	24	10	10	1⅛	24	16	10	2⅛
5/32	16	7	12	1⅛	16	10	8	1⅛	24	10	10	1⅛	24	16	10	2⅛
3/16	18	7	12	2⅛	18	10	8	2⅛	24	10	10	2⅛	24	16	10	2⅛
7/32	18	7	12	2⅛	18	10	8	2⅛	24	10	10	2⅛	32	20	16	3⅛
1/4	18	10	10	2⅛	24	10	10	2⅛	32	18	16	2⅛	36	22	18	3⅛
5/16	18	10	10	2⅛	24	10	10	2⅛	32	18	16	2⅛	36	22	18	3⅛
11/32	18	10	8	2⅛	32	16	16	2⅛	32	20	16	3⅛	50	16	26	3⅛
3/8	18	10	8	2⅛	32	16	16	2⅛	32	20	16	3⅛	50	16	26	3⅛

Dimensions given in inches

Round, solid, high-strength steel wire is to be shipped in coils or bundles protected with waterproof wrapping of paper.

In making racks for these reels allow 4 in. greater width than the traverse (*B*) specified.

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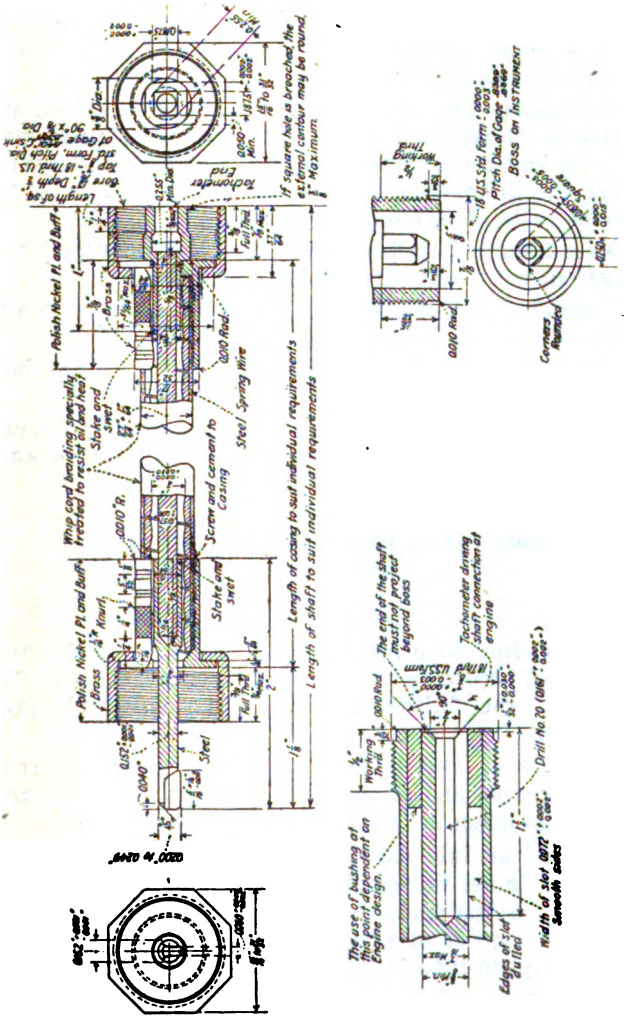
Tachometer Drive

Instrument End The upper or instrument end of the shaft has a squared socket, made integral with a shoulder, that forms a thrust bearing.

Engine End The lower or engine end is finished with a cylindrical end with a single integral key.

Casing The casing is to be of flexible metallic tubing,


Tachometer End
Length of casing to suit individual requirements
Length of shaft to suit individual requirements
Steel Spring Wire
Screw and cement to casing
Polish Nickel Pl and Buff
Brass
Steel
Boss on Instrument
Pitch Dia. of Gage
Working Thrd
Drill No 70
Edges of slot
Smooth edges
The end of the shaft must not project beyond boss
Tachometer driving shaft connection at engine
The use of bushing at this point dependent on Engine design.

Details of Tachometer Drive for Airplane Engines

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covered with whip-cord braid and impregnated to prevent oil leakage; similar to sample, or covered with other material that will be equally flexible, oil-tight, and free from stretching and of no greater weight than the whip-cord braid covering the casing. The outside diameter of casing to be over braid to be about 7/16 in.; the inside diameter of casing to be about ¼ in. The ends of the casing are finished alike, as shown in the accompanying sketches. The screw-threaded sockets covering the whip-cord braid are screwed in place and set with shellac or other suitable material to form an oil-tight and permanent connection.

Shaft-End The shaft-end at the engine is drilled and slotted to fit the tachometer shaft-end, allowing at least 1 in. for end-play. More play than this may be required where the drive-shaft is more than 6 ft. in length, the amount to be determined experimentally.

Drive-Shaft This is to be of helical wire-wound construction, with core wires.

Spindle The tachometer spindle is finished square to fit the square socket on the drive-shaft.

Drive-Shaft This will rotate at one-half crankshaft speed.

End-Nuts The size and contour of exterior circumference of end-nuts is optional, but a 15/16-in. octagon is recommended.

RUBBER HOSE FOR GASOLINE—TYPE NO. 1 (COTTON PLY)

Part I Technical

The gasoline hose must consist of rubber and cotton constructed according to the following specifications. The quality of the finished hose must be uniformly good throughout.

Inner Tube This is to be made of rubber free from flaws or cracks and of a quality conforming to the conditions laid down in this specification.

When of an internal diameter of ¾ in. or less, this inner tube must be seamless. Tubes of larger internal diameter may be made from rubber sheet, in which case they must consist of at least two turns, and be formed into a complete tube in the process of manufacture.

Cotton Plies The cotton shall consist of either braided or canvas plies which shall comply with the requirements given in the following table:

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Internal Dia., In.	Min. No. Plies
Up to and including 5/8	2
11/16 to 1½ inclusive	3
Above 1½	4

Outer Cover This is to be made of rubber of the same quality as that used for the inner tube, and of good workmanship and finish.

Dimension Limit The internal diameter of the hose must not vary more than plus or minus 3 per cent of the given internal diameter.

Length of Hose The manufacturer, when quoting or when acknowledging an order for this class of hose, must state the lengths in which it can be supplied. The hose should be supplied in the maximum lengths possible.

Physical Properties By "flexibility" is meant ability to bend without kinking. When hose of internal diameter (y) is bent around a cylinder having a diameter equal to x times the external diameter of the hose, as shown in the following table, the external diameter of the hose must not be increased or diminished by more than 10 per cent:

Internal Dia. (y) In.	x
Less than ½	8
½ to ¾	12
13/16 to 1⅛	14
Above 1⅛	16

The hose, after having been filled with gasoline for 2 hr., must withstand, without showing defects, a minimum internal hydraulic pressure (p), depending upon the internal diameter (d) of the hose.

Internal Dia. (d) In.	Min. Pressure (p) lb./sq. in.
Up to ½ inclusive	160
9/16 to 1 inclusive	140
1 1/16 to 1½ inclusive	120
1 9/16 to 2 inclusive	100
Above 2	80

Chemical Properties The rubber content shall not be less than 25 per cent by weight of plantation or wild hevea rubber. The amount of free sulphur in the tube, cover, or friction compounds shall not exceed 1 per cent.

Dry Heat Test A 3-in. piece of the hose, after having been placed in an air oven at 132 deg. cent. for 2 hr., must show, when cool, no tendency to crack, and must not be tacky.

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Permeability to Gasoline A 14-in. length of the hose is held vertically and plugged at the bottom. The upper end is fitted with a glass tube about 18 in. long. The hose so arranged is filled with gasoline* to a head of 12 in. above the top of the acting length of the hose. The acting length of the rubber hose is 12 inches. The upper end of the glass tube is loosely closed with a cork.

During the first 24 hr. the level of the gasoline will fall with comparative rapidity. The loss is made good by frequent additions from a known volume of gasoline, care being taken that the level of the gasoline in the glass tube does not fall by more than 3 in. at any time. The test is to last for 72 hr., and the loss of gasoline during the third 24 hr. must not exceed 100 cc. per square foot of the original internal surface of the hose.

Immersion in Gasoline A 3-in. piece of the hose is boiled for 1 hr. (using a reflex condenser) in gasoline similar to that used for the permeability test. The gasoline is allowed to cool down. Twenty-four hours later the test piece is removed from the gasoline and examined without delay, as follows:

The internal diameter at the point of greatest constriction is measured by means of rod gages. From this measurement the area of the bore is calculated. This must not differ from the original by more than 25 per cent.

The test-specimen is then cut longitudinally into halves, and the adhesion between rubber and cotton carefully examined. The adhesion must be of such a character that the rubber can only be stripped from the cotton by hand with difficulty.

Immersion in Oil A 3-in. piece of the hose is immersed in oil, approved by the purchaser, at a temperature of 100 deg. cent. for 8 hr., and for a further period of 24 hr. at ordinary temperature. The oil is then wiped from the surface of the hose. The decrease of internal diameter shall be less than 10 per cent.

The flexibility and elasticity of the rubber must not be diminished and there must be no tendency of the rubber to separate from the cotton.

Part II Inspection

Test-Specimens The purchaser will decide where the tests are to be carried out. All test-specimens are to be

*The specific gravity of the gasoline used in this test should be between 0.710 and 0.725 at 15.5 deg. cent. (60 deg. fahr.); 65 per cent of it must distill over at 100 deg. cent. (212 deg. fahr.) from a distillation flask when the bulb of the thermometer is just below the side tube

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cut in the presence of the inspector and are to be marked as he may direct.

For the purpose of testing, a representative sample will be cut from each 1000 ft. of hose or fraction thereof, and the tests will proceed in accordance with the purchaser's instructions.

Rejections If any sample fails to comply with any of the above tests, the hose represented thereby will be rejected.

Marking Accepted and rejected material must be marked as directed by the inspector.

Part III Special Conditions

Depreciation The manufacturer must bear the cost of the depreciation in value of any rejected material due to test specimens being cut therefrom.

Rejected Material The manufacturer must not supply any material that has previously been rejected by any Government Department, without giving full written particulars of the previous rejection to the inspector who is detailed to this specification.

Rubber Hose for Gasoline—Type No. 2
(cotton ply reinforced with wire)

Part I Technical

The gasoline hose must consist of rubber and cotton constructed according to the following specifications. The quality of the finished hose must be uniformly good throughout.

Inner Tube This shall consist of a canvas tube supported by a helix of oil-tempered steel wire of not less than 0.020 in. diameter, spaced not less than five turns per inch. The rubber tube shall be placed between this inner cotton tube and the cotton plies.

Cotton Plies The cotton shall consist of either braided or canvas plies, which shall comply with the requirements of the following table. The inner cotton tube shall not be considered one of the plies specified.

Internal Dia., In.	Min. No. Plies
Up to and including 5/8.....	2
11/16 to 1½ inclusive.....	3
Above 1½.....	4

Outer Cover This is to be made of rubber of the same

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quality as that used for the inner tube, and of good workmanship and finish.

Dimension Limit The internal diameter of the hose must not vary more than plus or minus 3 per cent of the given internal diameter.

Length of Hose The manufacturer, when quoting or when acknowledging an order for this class of hose, must state the lengths in which it can be supplied. The hose should be supplied in the maximum lengths possible.

Physical Properties By "flexibility" is meant ability to bend without kinking. When hose of internal diameter (y) is bent around a cylinder having a diameter equal to x times the external diameter of the hose, as shown in the following table, the external diameter of the hose must not increase or diminish by more than 10 per cent.

Internal Dia. (y), In.	x
Less than ½	8
½ to ¾	12
13/16 to 1⅛	14
Above 1⅛	16

The hose, after having been filled with gasoline for two hours, must withstand a minimum internal hydraulic pressure of (p) pounds per square inch, depending upon the internal diameter (d) of the hose without showing defects.

Internal Dia. (d) In.	Min. Pressure (p) lb./sq. in.
Up to ½ inclusive	160
9/16 to 1 inclusive	140
1 1/16 to 1½ inclusive	120
1 9/16 to 2 inclusive	100
Above 2	80

Chemical Properties The rubber contents shall not be less than 25 per cent by weight of plantation or wild hevea rubber. The amount of free sulphur in either the tube, cover, or friction compounds shall not exceed 1 per cent.

Dry Heat Test A 3-in. piece of the hose after having been placed in an air oven at 132 deg. cent. for 2 hr. must show, when cool, no tendency to crack, and must not be tacky.

Permeability to Gasoline A 142-in. length of hose is held vertically and plugged at the bottom. The upper end is fitted with a glass tube about 18 in. long. The hose so arranged is filled with gasoline* to a head of 12 in.

*The specific gravity of the gasoline used in this test should be between 0.710 and 0.725 at 15.5 deg. cent. (60 deg. fahr.); 65 per cent of it must distill over at 100 deg. cent. (212 deg. fahr.) from a distillation flask when the bulb of the thermometer is just below the side tube

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above the top of the acting length of the hose. The acting length of the rubber hose is 12 in. The upper end of the glass tube is loosely closed with a cork.

During the first 24 hr. the level of the gasoline will fall comparatively rapidly. The loss is made good by frequent additions from a known volume of gasoline, care being taken that the level of the gasoline in the glass tube does not fall by more than 3 in. at any time. The test is to last for 72 hr., and the loss of gasoline during the third 24 hr. must not exceed 100 cc. per square foot of the original internal surface of the hose.

Immersion in Gasoline A 3-in. piece of the hose is boiled for 1 hr. (using a reflex condenser) in gasoline similar to that used for the permeability test. The gasoline is allowed to cool down. Twenty-four hours later the test piece is removed from the gasoline and examined without delay, as follows:

The internal diameter at the point of greatest constriction is measured by means of rod gages. From this measurement the area of the bore is calculated. It must not differ from the original by more than 25 per cent.

The test-specimen is then cut longitudinally into halves, and the adhesion between rubber and cotton carefully examined. The adhesion must be of such a character that the rubber can only be stripped from the cotton by hand with difficulty.

Immersion in Oil A 3-in. piece of the hose is immersed in oil approved by the purchaser at a temperature of 100 deg. cent. for 8 hr., and for a further period of 24 hr. at ordinary temperature. The oil is then wiped from the surface of the hose. The decrease of internal diameter shall be less than 10 per cent.

The flexibility and elasticity of the rubber must not be diminished and there must be no tendency of the rubber to separate from the cotton.

Part II Inspection

Test-Specimens The purchaser will decide where the tests are to be carried out. All test-specimens are to be cut in the presence of the inspector and they are to be marked as he may direct.

For the purpose of testing, a representative sample will be cut from each 1000 ft. of hose or fraction thereof, and the tests will proceed in accordance with the purchaser's instructions.

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Rejections If any sample fails to comply with any of the above tests, the hose represented thereby will be rejected.

Marking Accepted and rejected material must be marked as directed by the inspector.

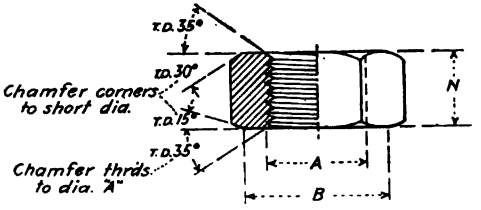
Part III Special Conditions

Depreciation The manufacturer must bear the cost of the depreciation in value of any rejected material due to test-specimens being cut therefrom.

Rejected Material The manufacturer must not supply any material that has previously been rejected by any Government Department, without giving full written particulars of the previous rejection to the inspector who is inspecting to this specification.

Plain Hexagon Nuts

(For bodies and wings, not engines)



T.D. = Tooling Dimensions

Dimensions for Plain Hexagon Nuts

Size *A*	Threads per In.	Hex. Short Dia. *B*	*N*
*0.1120 (No. 4)	To be	0.250 (¼)	0.094 (3/32)
*0.1380 (No. 6)	determined	0.312 (5/16)	0.109 (7/64)
0.1640 (No. 8)	32	0.375 (⅜)	0.125 (⅛)
0.1900 (No. 10)	32	0.375 (⅜)	0.141 (9/64)
0.2160 (No. 12)	32	0.438 (7/16)	0.156 (5/32)
0.2500 (¼)	28	0.438 (7/16)	0.188 (3/16)
0.3125 (5/16)	24	0.500 (½)	0.234 (15/64)
0.3750 (⅜)	24	0.563 (9/16)	0.281 (9/32)
0.4375 (7/16)	20	0.688 (11/16)	0.328 (21/64)
0.5000 (½)	20	0.750 (¾)	0.375 (⅜)
0.5625 (9/16)	18	0.875 (⅞)	0.422 (27/64)
0.6250 (⅝)	18	0.938 (15/16)	0.469 (15/32)

All Threads U. S. Form. *Future Sizes

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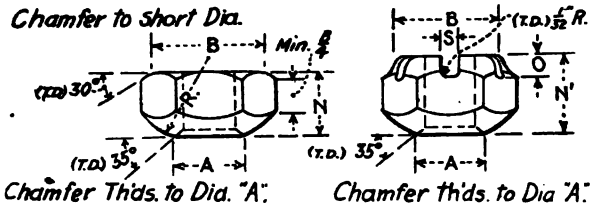
Ball Hexagon and Castle Nuts

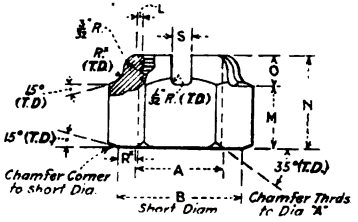


$N^1 = N + O$. T.D. = Tooling Dimensions

Bolt Size *A*	Thds. per Inch	Hexagon			Castle			
		B	*N*	*R*	*B*	N^1	*S*	*O*
0.5625 (9/16)	18	0.875 (7/8)	0.469 (15/32)	0.625	0.875 (7/8)	0.656 (21/32)	0.156	0.188 (3/16)
0.6250 (5/8)	18	0.938 (15/16)	0.516 (33/64)	0.625 (5/8)	0.938 (15/16)	0.766 (49/64)	0.156 (5/32)	0.250 (1/4)

All Threads U. S. Form *B* = Also Size of Hexagon

Castle Hexagon Nuts



T.D. = Tooling Dimensions

Bolt Size *A*	Thds. per Inch	*B*	*N*	*S*	*O*	*M*	*L	*R*
0.5625 (9/16)	18	0.875 (7/8)	0.609 (39/64)	0.156	0.188 (3/16)	0.422 (27/64)	0.047	0.156
0.6250 (5/8)	18	0.938 (15/16)	0.719 (23/32)	0.156 (5/32)	0.250 (1/4)	0.469 (15/32)	0.047 (3/64)	0.156 (5/32)

All Threads U. S. Form *B* = Also Size of Hexagon *O* = Also Depth of Slot

*L = Distance from A to beginning of 3/32 in. radius whose center is floating to satisfy curve

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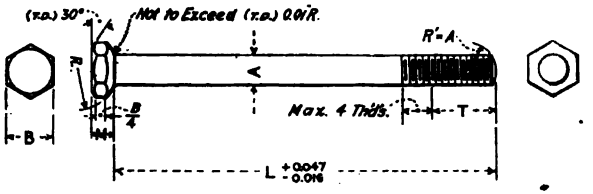
Ball Hexagon-Head Bolts



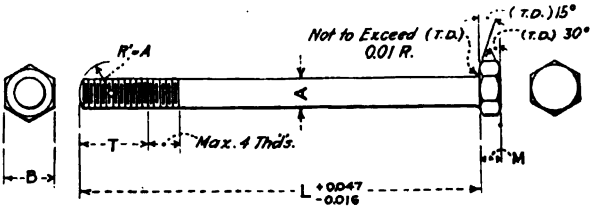
T = Min. Length of Usable Thread. T.D. = Tooling Dimensions

Bolts Size *A*	Threads per Inch	*A* *Limits	*B*	*M*	*R*
0.1640 (No.8.)	32	0.1640 0.1600	0.313 (5/16)	0.172 (11/64)	0.375 (3/8)
0.5625 (9/16)	18	0.5625 0.5580	0.875 (7/8)	0.469 (15/32)	0.625 (5/8)
0.6250 (5/8)	18	0.6250 0.6200	0.938 (15/16)	0.516 (33/64)	0.625 (5/8)

Body length *L* =	1/2	5/8	3/4	7/8	1	Over 1
Length of thread *T* =	3/8	7/16	1/2	1/2	1/2	3/4

All threads U. S. Form. All dimensions in inches.
*Finished size, including plating or rust-preventing treatment when used. The Bureau of Standards recommends 0.001 in. as thickness of plating for rust prevention

Plain Hexagon-Head Bolts



T = Min. Length of Usable Thread. T. D. = Tooling Dimension

Bolts Size *A*	Threads per Inch	*Limits *A*	*B*	*M*
0.563 (9/16)	18	0.5625 0.5580	0.875 (7/8)	0.313 (5/16)
0.625 (5/8)	18	0.6250 0.6200	0.938 (15/16)	0.344 (11/32)

Body length *L* =	1/2	5/8	3/4	7/8	1	Over 1
Length of thread *T* =	3/8	7/16	1/2	1/2	1/2	3/4

All threads U. S. Form. $M = (A/2 + 1/32)$ in. All dimensions in inches.
*Finished size including plating or rust-preventing treatment when used. Bureau of Standards recommends 0.001 in. as thickness of plating for rust prevention

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Clamps and Fittings for Rubber Hose

In order to comply with present practice and utilize existing hose connections, the list of sizes* now standardized by the Society are recommended for aeronautic practice.

Airplane Engine Testing

The recommendation on engine testing was adopted by the Society, but in the final letter ballot several negative votes were accompanied by comments indicating that developments in the art had rendered the proposal obsolete, even in the short interim between its formulation and adoption. The Aeronautic Division has therefore taken the subject under consideration in order to revise its report in accordance with up-to-date practice.

The following test specifications are divided into two classes, one a Routine Factory Test and the other a Type Test.

Routine Factory Test for Fixed Engines

(Low or Normal Compression)

Test No. 1 Engines may be run in by such methods as are found most satisfactory for each type of engine, but must not be run under own power over ten hours before the first official test.

Test No. 2 Engines with compression ratio of 5 or less.

Engine will be mounted on a torque stand or stationary stand and fitted with club or propeller, properly calibrated for rated load at rated speed. Engine will be fitted with all the accessories that are to be supplied by the engine manufacturer. If started by hand, it must in every attempt be started within 5 min. by one man.

Test will consist of four hours' continuous running at rated power and rated speed. All readings required shall be recorded on S. A. E. standard log sheet every ten minutes.

(a) The test must be continuous and in case of a stop occurring during the first hour, the run will be repeated.

(b) In case of a stop occurring after the first hour and not due to a fault of the engine, the run will be continued, but the inspector may require an extension of the running time equal to twice the time lost.

*See S. A. E. Transactions, Part I. p. 17. or S. A. E. Handbook. Vol. I, sh. 37

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When a stop occurs at any time due to a fault of the engine, the inspector may require the complete run to be repeated.

(c) No manipulation of the engine, while running, will be allowed, except such as can be made by a pilot from his seat. Replenishing of gasoline and oil supplies and regulating of the cooling water temperature are allowable.

Engine will be dis-assembled, inspected, and if all parts are accepted, it will be re-assembled. If in the opinion of the inspector, parts must be replaced, he may require the whole or a portion of the run to be repeated.

Test No. 3 Engine with all the accessories that are to be supplied by the manufacturer will be mounted on a torque stand and fitted with a club or propeller, properly calibrated. It will be given a continuous run of one hour at not less than rated power and rated speed. All readings required shall be recorded on S. A. E. standard log sheet every ten minutes.

After this test the engine shall be cleaned and inspected and put into condition for shipment. If satisfactory, the engine shall be stamped by the inspector.

Note—If desired, the horsepower may be corrected for air temperature and barometric pressure. In such cases the power will be corrected to a surrounding air temperature of 59 deg. fahr. (15 deg. cent.) and normal sea level pressure which is 29.92 in. (760 mm.) of mercury at 32 deg. fahr. (0 deg. cent.) and at 45 deg. latitude. In case of question the corrected power shall govern.

(High Compression)

Test No. 2*A* Engines with compression ratio of 5 or more.

Engine will be prepared as under Test No. 1 and No. 2.

This test will consist of four hours' continuous running at 8/10 rated load and rated speed. Otherwise this test will be the same as No. 2 and the same rules will apply.

Dis-assembly and inspection will be the same as in Tests No. 2 and No. 3.

Test No. 3*A* This will be the same as No. 3 but at 8/10 rated load and at rated speed.

After this test, the engine will be inspected and accepted as in Routine Test No. 3.

Type Tests for Fixed Engines

(Low or Normal Compression)

The first engine of a type and one out of each hundred in production will be tested under the following conditions:

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The selection of the engine shall be made by the inspector without previous notice to the manufacturer.

The engine must not run under its own power over ten hours before the first official test and must be taken from regular production. (See Routine Test No. 1)

Test No. 4 Engines with compression ratio of 5 or less.

The engine is to be mounted and equipped as in Routine Test No. 2 and fitted with a club or propeller properly calibrated.

Total running time shall be fifty hours divided as follows:

Five hours at 9/10 rated power and at rated speed.

(Not over one hour's stop with minor adjustments allowed)

Five hours at 9/10 rated power and at rated speed.

(Not over 13 hours' stop during which engine may be dis-assembled, overhauled, carbon removed, valves ground and adjusted, magnetos and spark-plugs cleaned and adjusted. No adjustment or replacement of major parts is allowed. Any multiple part, such as a valve, may be replaced when replacement is made necessary through defective material. In no case, however, shall more than one such part be replaced during the entire fifty hours)

Five hours at 9/10 rated power and at rated speed.

(Not over one hour's stop with minor adjustments allowed)

Five hours at 9/10 rated power and at rated speed.

(Not over 13 hours' stop and same work allowed as before)

Five hours at 9/10 rated power and at rated speed.

(Not over one hour stop with minor adjustments allowed)

Five hours at rated speed and at rated power.

(Not over 13 hours' stop and same work allowed)

Five hours at rated speed and at rated power.

(Not over one hour's stop with minor adjustments)

Five hours at rated speed and at rated power.

(Not over 13 hours' stop and same work allowed)

Ten hours at rated power and at rated speed.

Rule (c) under Routine Test No. 2 will apply to each run. In case of a forced stop during any of above runs, rules (a) and (b) under Routine Test No. 2 will apply.

If the stop is due to a failure of any multiple part such as a valve, the part may be replaced, provided such failure is due to defective material and a similar part

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has not previously been replaced since the 50-hr. test was started.

During complete run all required readings must be recorded on S. A. E. standard log sheet every ten minutes The mean power (see note under Routine Test No. 3) must not fall below the rated power; and gasoline and oil consumption must not exceed the predetermined maximum.

After the test is completed, the engine is to be dis-assembled and if acceptable to the inspector, may be reassembled after replacing any parts that the inspector may designate.

Test No. 5 The engine is to be mounted in a body (fuselage) or tilting stand and fitted with propeller. With the axis of the engine making 15 deg. with horizontal, it is to run at full power and full speed for fifteen minutes. Then without altering speed or any part, or adjustment, the axis is to be brought again horizontal and then tilted to 15 deg. in the opposite direction and run fifteen minutes.

If this test is satisfactory, the engine will be put through Routine Test No. 3 except that the running time shall be thirty minutes instead of one hour. After this test, the engine will be inspected and accepted as in Routine Test No. 3.

(High Compression)

Test No. 4*A* Engines with compression ratio of 5 or more.

This test will be the same as test No. 4 but the first five runs shall be made at 7/10 rated load and at rated speed. The next four runs will be made at 8/10 rated load and at rated speed.

Dis-assembly and inspection will be the same as in Tests No. 4 and No. 5.

Test 5*A* Engine will be put through a test the same as No. 5 but at 8/10 rated load and at rated speed.

If the engine passes Test 5A it will be put through Routine Test No. 3A except that the running time will be thirty minutes instead of one hour. This engine will then be inspected and accepted as in Routine Test No. 3.

Airplane Engine Testing Form

Name and Model..... Date of Test.....
Manufacturer .....

(All dimensions to be given in inches)

(1) General Type..... Cycle .....
(2) No. of Cycle..... Bore ..... Strol .....

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Piston Displ. per Cyl.....cu. in. Total.....cu. in.

(3) Compression Vol. (V_c).....cu. in.
Total Vol. of Cyl. (V).....cu. in.
Compression Ratio = ($V_c \div V$)....Compression Pressure .....lb. At.....r.p.m.

(4) Type of Cyl..... Mat'l.....

(5) Type of Valves..... Location.....

(6) Cooling System..... Type of Pump.....
Capacity.....lb. per min. at.....eng. r.p.m.
Weight of Water in Cyl. Jackets.....lb.
Water Space bet. Jacket and Cyl., Top.. Bottom..

(7) Piston, Type..... Mat'l.....
Weight without Pin and Rings..... With Pin and Rings.....
Distance from center of Pin to top of Piston.....
Length, over all.. Clearance Main Body.. Top..
Length Piston Clearance at Head.....
Length Relieved Portion at Center if any.....

(8) Piston Rings, No. per Piston.. Type.. Width...

(9) Connecting-Rod Type..... Mat'l.....
Length, c. to c..... Weight, Upper end.....
Lower end..... Total.....
No. Bolts, Lower End..... Dia. Bolts.....
Mat'l..... Weight complete with Bearing.....

(10) Piston-Pin Bearings, dia... Length... Mat'l...
Location..... Method of fastening Pin in Rod or Piston..... Weight of Pin.....

(11) Connecting-Rod Bearing, dia.. Length.. Mat'l..
Type.... Weight.... Method of Adjustment....
Method of Holding Bearing in Rod.....

(12) Crankshaft, Type..... Weight.....
No. of Bearings.. Dia. Main.. Length of Front..
Length, Intermediate..... Length, Rear.....
Method of Adjusting Bearings.....
No. of Bolts each Bearing... How held in Case...
Dia. Hole in Main Bearings.... In Crankpins....
Dia. Hole in Crankpins..... Dia. Hole, Main Bearings.....
Cheek Thickness next to Propeller End.....
Cheek Thickness of Balance of Crank.....
Dia. of Shaft through Radial Bearing.....
No. of Radial Bearings..... No. of Thrust Bearings.....
Type of Thrust Bearing.. If Direct, what Type..
Propeller Mounting.. Length of Taper.. Dia...
Method of Holding Propeller Hub.....

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No. of Hubs..... If Geared, how is Gear held to Shaft.....
If Taper, give Length.....in. Dia.....in.
Is Radial used in addition to Plain Bearing.....
(13) Camshaft Bearings, No..... Dia.....in.
Material..... Lengths.....in.
(14) Type of Cams..... Type of Valve-Lifters.....
(15) Inlet Valves, No. per Cyl.....o.d.....in. Port dia.....in.
Lift.....in. Seat Angle.....deg.
(16) Exhaust Valves, No. per cyl.... o.d.....in. Port dia....in. Lift....in. Seat angle....deg.
(17) Weight of Valve Reciprocating Parts, Inlet....lb. Exhaust.....lb.
(18) Valve-Spring Tension, Inlet open.....lb. Closed .....lb.
Exhaust open.....lb. Closed.....lb.
*(19) Valve-Timing, Inlet opens.....deg. before (after) Top Center
Closes.....deg. after Lower Center
Exhaust opens.....deg. before Lower Center
*Closes.....deg. before (after) Top Center
(20) Geared Drive, Pitch..... No. Teeth Crankshaft Gear.....
No. Teeth Propeller Gear.... Width of Face....
Type Gear..... Mat'l.....
(21) Propeller-Shaft, Type Thrust Bearing.....
No..... Radial Bearing, Front..... Rear.....
Type of Propeller-Hub Mounting...No. of Hub...

Carburetion

(22) Carbureter, Name..... Model.....
(23) Specification (size of nozzles, etc.).....
(24) How Heated.....
(25) General Principles of Operation.....
(26) Description of Intake Pipe.....

Ignition

(27) Name..... System..... Type.....
(28) Type of Distributor..... Firing Order.....
(29) Type of Breaker.....
Maximum Spark-Advance...deg. Retard....deg.
(30) Spark-Plugs, Name..... Type.....
No..... Size.....

Lubrication

(32) Type..... Description.....
Pressure used.....
(33) Main Bearing Hub.....
(34) Crankpin.....
(35) Cylinder.....

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(36) Method of Oil Cooling.....
(37) Oil Container..... Capacity.....
(38) Pump..... Type.....
(39) Oil used.....

Engine Weight Specifications

Manufacturer.....
Model.....
Bore.....in.....mm. Stroke.....in.....mm.
No. of cyl..... Type.....
Piston displacement.....cu. in.....liters
Compression ratio.....
Horsepower.....at.....r.p.m. Computed at standard barometric pressure 29.92 in. (760 mm.), 32 deg. fahr. (0 deg. cent.), 45-deg. latitude at sea level.
Gasoline consumption....lb....kilo. per hr. at above hp.
Oil consumption.....lb....kilo. per hr. at above hp.
Propeller speed.....r.p.m. at above engine speed
Weight of engine—dry.....lb.....kilo.

The weight of the engine includes the following:

1 IGNITION Service magneto. Wire and wire supports. (or) Battery ignition distributor and coils, wire and wire supports. Spark-plugs.

2 GENERATOR (if used) attached. With cut-out and ammeter.

6 CARBURETERS

7 INLET PIPES

Note.—Numbers refer to parts given in following schedule: Comparison of standard water-cooled engines, with rotary and fixed air-cooled types, may be made by means of schedule of weights. Give all weights in the following schedule in the proper column, and give names and weights of any other necessary parts not listed.

Mark X if No Used	Equipment	Manufactured by	WEIGHTS			
			Included in Above Engine Wt.		Not Included in Above Engine Wt.	
			Lb.	Kilo.	Lb.	Kilo.
	1 Ignition					
	a–Service magneto (No. required.....), with attaching bolts or clamps, and complete drive.....					
	Wire and wire supports..					
	Switch.....					
	b–Starting magneto, with attaching bolts or clamps, and complete drive.....					
	Wire and wire supports..					
	Switch.....					

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Mark X if Not Used	Equipment	Manufactured by	WEIGHTS			
			Included in Above Engine Wt.		Not Included in Above Engine Wt.	
			Lb.	Kilo.	Lb.	Kilo.
	c–Battery ignition distributor (No. required.....), with bolts or clamps and complete drive					
	Coil (No. required,.....)					
	Wire with wire supports					
	Switch					
	d–Battery					
	(Capacity.....amp. hr.)					
	(Voltage.....)					
	Box, hold-down clamp and cover					
	e–Spark-plugs (No. required .....)					
	2 Generator					
	(Type of regulation.....)					
	a–Complete with cut-out, regulator, drive, bolts or clamps					
	Is ignition distributor included?					
	b–Ammeter (or voltmeter)					
	3 Electric Starter					
	a–Motor with gearshift drive, mounting brackets, strap, bolts and terminals (starting wire on body)					
	b–Starting battery					
	(Capacity.....amp. hr.)					
	(Voltage.....)					
	Terminals					
	Switch					
	Box and hold-down clamps					
	4 Compressed Air Starter					
	a–Starter and compressor complete with gearshift, drive, and all piping except between starter and tank					
	b–Air storage tank with straps					
	c–Control valve					
	d–Pipe couplings (piping on body)					
	5 Hand-Starting Cranks Including attaching bolts and nuts					
	6 Carbureters (No. required.....), with attaching bolts and nuts					
	7 Inlet Pipes With bolts and nuts					
	8 Hot-Air Pipes or Stoves					
	9 Exhaust Pipes With flanges and attaching bolts and nuts					

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Mark X if Not Used	Equipment	Manufactured by	WEIGHTS			
			Included in Above Engine Wt.		Not Included in Above Engine Wt.	
			Lb.	Kilo.	Lb.	Kilo.
	10 Mufflers (No. required.....)					
	11 Propeller-Hub Complete with bolts and nuts					
	12 Gasoline Air-Pressure Pump With drive, bolts and nuts (piping on body)					
	13 Gasoline Pump With drive-bolts and nuts (piping on body)					
	14 Tachometer Gear Drive on Engine					
	15 Machine Gun Timing Mechanism					
	16 Water Radiator (No. required.....) Water radiator connections to engine manifolds					
	17 Oil Radiator or Tank With connections					
	18 Oil Capacity of Crankcase					
	19 Water Capacity of Jackets, Pumps and Manifolds					
	20 Water Capacity of Radiators					
	21					
	22					
	23					
	24					
	25					

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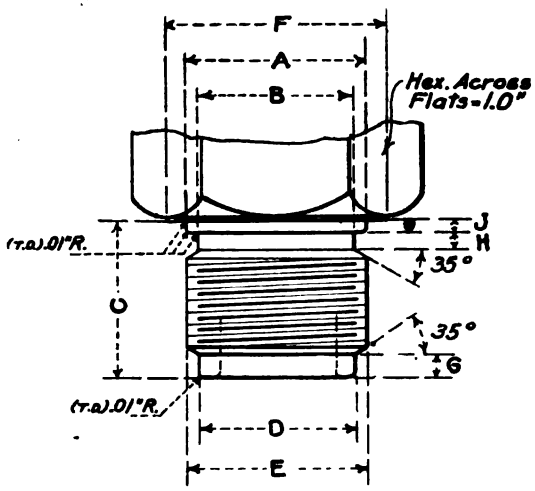
Spark-Plug Shell Dimensions

The thread is to be 18 mm. with 1½ mm. pitch.

The form of thread should be the International standard (same as U. S. standard, except that truncation is one-half as much at root of thread).



Spark-Plug Shell Dimensions

Dimension	Maximum		Minimum	
	Mm.	In.	Mm.	In.
A	18.03	0.710	17.93	0.706
B	16.87	0.664	15.87	0.625
C	15.87	0.625 (⅝)	15.87	0.625 (⅝)
D	15.87	0.625	.....	.....
E	17.97	0.708	17.85	0.703
F	.....	.....	23.83	0.938 (15/16)
G	.....	.....	2.38	0.094 (3/32)
H	1.98	0.078 (5/64)	.....	.....
J	1.19	0.047 (3/64)	.....	.....

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SPARK-PLUG THREAD DIMENSIONS

Diameter	MAXIMUM		MINIMUM	
	Mm.	In.	Mm.	In.
Outside.....	17.975	0.70768 (0.708)	17.850	0.70275 (0.703)
Pitch.....	17.001	0.66933 (0.669)	16.876	0.66441 (0.664)
Root.....	15.864	0.62457 (0.625)	15.739	0.61964 (0.620)

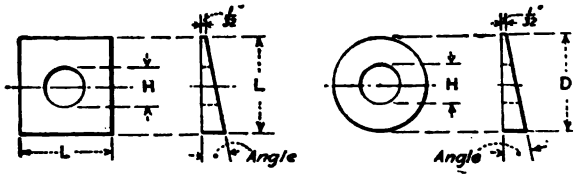
TAPPED HOLE DIMENSIONS

Diameter	MAXIMUM		MINIMUM	
	Mm.	In.	Mm.	In.
Outside.....	18.325	0.72146 (0.721)	18.187	0.71603 (0.716)
Pitch.....	17.176	0.67622 (0.676)	17.051	0.67124 (0.671)
Root.....	16.201	0.63783 (0.638)	16.076	0.63291 (0.633)

The limits for spark-plug threads as shown in the accompanying table have been reported as adopted by the (British) Engineering Standards Committee

The standard thread for spark-plug terminal is No. 8-32 (0.164 dia.), A. S. M. E. standard

BEVEL WASHERS



DIMENSIONS OF SQUARE AND ROUND BEVEL WASHERS

Bolt Size*	*H*	*D*	*L*
No. 4 (0.112)	No. 31 (0.120)	3/8	3/8
No. 6 (0.138)	No. 26 (0.147)	7/16	3/8
No. 8 (0.164)	No. 17 (0.173)	1 9/32	1 9/32
No. 10 (0.190)	No. 8 (0.199)	1/2	1 9/32
No. 12 (0.216)	No. 1 (0.228)	9/16	1 17/32
1/4	1 7/64	5/8	1 17/32
5/16	2 1/64	1 1/16	2 1/32
3/8	2 1/64	3/4	2 1/32
7/16	2 1/64	1 5/16	7/8
1/2	2 1/64	1	7/8

*The values of the angles are 6, 12, 24 and 36 deg. for all sizes

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DISCUSSION AT STANDARDS COMMITTEE MEETING

C. M. MANLY (M. S. A. E.):—So much change is likely to develop in aeronautical work, owing to the rapid advancement of the art, and the many things have had to be codified, that it seemed unwise to adopt standards which might have some flaws in minor or even in major particulars and soon afterward have to withdraw them. We therefore recommended that the specifications be adopted as S. A. E. Recommended Aeronautic Practice instead of as Standard.

Types of Dopes

The recommendations for both the cellulose acetate and nitrate dopes have been studied by a subdivision headed by Doctor Smith of the Bureau of Standards. I will ask him to comment on them.

DR. W. H. SMITH: The specifications are not ideal, but they are practical. The Signal Corps is drawing up specifications for its own use, taking these as a basis and incorporating certain changes which are necessary, owing to the organization of an Inspection Department. We hope later to incorporate a baking test for the determination of durability.

These specifications will also be used by the International Aircraft Standards Board in drawing up specifications on dope. One important change which the Signal Corps has made is the absolute exclusion of tetrachlorethane from dopes.

Round High Strength Steel Wire

MR. MANLY: Two different committees worked on the drawing up of the specifications for round, high-strength steel wire for aeronautics and later on, the International Aircraft Standards Board. At the time the original specifications were brought out the sizes of wire manufactured were limited in number, but since then they have been extended materially to include both smaller and larger sizes. The original torsion test was so severe for the larger sizes that it was impossible to get them to pass it.

STANDARDS MANAGER M. W. HANKS (M. S. A. E.): The original torsion test specification called for the number of turns to equal 1.1 divided by the diameter of the wire. When the International Aircraft Standards Board took up the subject it changed the specification so that it was 2.7 instead of 1.1, which, of course, permits a greater number of turns.

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JOHN V. COSTELLO (M. S. A. E.):—It is possible that something in this specification will be changed by the Signal Corps. It would be better before adopting it to arrange for some one to work with representatives of the Signal Corps.

E. T. BIRDSALL (M. S. A. E.):—It seems to me that different sizes of wire require different numbers of turns. It is a question of fiber stress on the surface.

MR. MANLY:—The only question is whether it should be 2.7 divided by the diameter in inches or some other particular decimal.

Non-Flexible 19 Steel Wire Cable

MR. MANLY:—Mr. Horn of the John A. Roebling's Sons Co. states that non-flexible steel wire cable is referred to as galvanized wire in the United States Government specifications. The Government insists on having the wire tinned. Mr. Horn contends that galvanized wire is really superior to tin-coated wire and that specifying tin in place of galvanized wire would really not give as high a grade of material. Some of the members of the International Aircraft Standards Board insisted on having tin rather than galvanizing.

F. G. ERICSON (M. S. A. E.):—Galvanizing is not as good a coating on the wire. None of the foreign representatives would agree to adopt the galvanized wire.

MR. MANLY:—On the other hand, the United States Government would not use the tinned wire. The Roebling Company is now called upon to make both. So long as the different Governments are in disagreement on this subject I think the Society ought to take the middle ground and recommend both.

MR. BIRDSALL:—I know very little concerning the application of airplanes but do know about shipping. We have always believed that galvanizing weakens a wire cable or a chain cable, and I think tests will show that when the wire rigging of ships is galvanized the strength is much less than that of plain steel wire. Galvanizing weakens a wire, a rod or a chain.

MR. MANLY:—Is that hot, or electro-galvanizing?

MR. BIRDSALL:—Hot galvanizing; it seems to have an effect on the surface of the steel.

T. B. FARNSWORTH (M. S. A. E.):—I suggest that the words "hot galvanized" be added. If the wire is electro-galvanized, it may not be accepted. Why not let the manufacturer know at the start what he is to produce?

MR. MANLY:—We had that same question up in connection with thimbles. The Navy Department wanted

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them hot galvanized but we left the specifications to read "galvanized."

MR. FARNSWORTH:—It is true that there is an electrolytic action on any galvanized surface exposed to salt air. I would specify hot galvanizing in the beginning because there is no question about that being the best.

MR. MANLEY:—That may be true in the navy, but not in the army. I think we had better leave just the word "galvanized."

(The amendment to insert the words "or galvanized" was carried)

MR. BIRDSALL:—I suggest adding a footnote stating that the customer shall specify whether hot or electro-galvanizing is to be employed.

MR. MANLY:—I want to call the manufacturers' attention to the fact that there is a preference in connection with the application of galvanizing and that everybody is not agreed as to the method. I am willing to have the footnote read, "The method of galvanizing, if used, is to be specified by the customer."

(It was voted to add this footnote)

Extra Flexible 7 by 19 Steel Wire Cable

MR. ERICSON:—It is not specified exactly how this 7 by 19 wire cable is to be made. The wire made by Roebling to this specification is of peculiar construction, consisting of three different sizes of wire. Some manufacturers hold that to be the reason why this cable has such high strength. The English 7 by 19 wire cable is of 30 per cent less strength owing to the difference in its construction. It is not quite as flexible as the regular wire.

MR. MANLY:—We are not particularly interested in construction so long as the desired strength and degree of flexibility are obtained. If possible, I would rather specify the degree of flexibility than tell the manufacturer how to make his cable.

I would suggest that we change the wording of the second paragraph under the section "Manufacture" from, "It is to be understood that the strand composing this cable must necessarily be composed of wires of the same diameter," to read: "It is to be understood that the strand composing this cable must not necessarily be composed of wires all of the same diameter."

(It was voted to accept this amendment)

MR. BIRDSALL:—I do not find any test for elongation on these stranded cables. Stranded cables have considerable elongation under strain and those with cotton or hemp cores have still more elongation. With the varia-

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tion of core and of lay there might be a great variation in the strength of standard cable when strained.

MR. MANLY:—The elongation of the material is taken care of by the torsion and bending tests. There has been no specification covering the amount of permissible elongation on the cable test; this is due to the fact that all this cable is apt to be stretched to 50 per cent of the breaking point before it is applied. That is supposed to take the initial stress out of the cable. Of course, it would affect the user if he were getting cables of various degrees of "stretchability." If the wire manufacturer should supply wire which gives trouble in that way, he would hear from the airplane manufacturer without delay. I think it is better to leave this question to the wire manufacturers.

Reels for Cable

The cable reel specifications were prepared to facilitate stockroom storage. Dimensions are immaterial, so long as those given in the table are not exceeded. If it is desired to lighten these reels it would not make any difference as long as they have the core and outside diameters and the specified traverse, so that they can be stored in the same stockroom space.

Rubber Hose for Gasoline

J. C. TUTTLE (M. S. A. E.):—Under the heading, "Chemical Properties," the first paragraph reads: "No organic matter other than Para or plantation rubber shall be used in the preparation of this hose." Some manufacturers have secured an excellent product by the use of material other than rubber, therefore I suggest that this paragraph be changed to read: "If organic matter other than Para or plantation rubber be used, the amount and the identity of such material shall be stated." Then if a manufacturer proposes to use something that we know to be poor, we can take care of it before anything is done on production.

In the next sentence is the statement, "The percentage of rubber shall not be less than 32." The Subdivision that prepared the specification is unanimous in desiring that this be changed to read, "The percentage of rubber shall not be less than 25."

J. E. HALE (M. S. A. E.):—It is customary for the Government departments to require that in certain cases the material used in a particular product be specifically named. I do not think it is necessary to handle anything in that way in the S. A. E. specifications, and, further-

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more, there will be trouble in getting this information.

The physical tests on this tubing are rigid, and I know positively that if any tubing can pass the gasoline tests, and the flexibility, structure and aging tests, it will be amply safeguarded in quality. For these reasons I object to requiring rubber manufacturers to disclose the nature of any organic substances used.

MR. MANLY:—I do not know that we want to require a manufacturer to disclose trade practices in connection with his material, provided it meets the tests specified and provided those tests do cover the securing of a suitable result in the form of hose.

MR. TUTTLE:—I object to leaving the composition of the material so open that the manufacturer is free to do just as he pleases, because all tests are made on the finished article and do not cover service. The manufacturer may use material that will meet every test we can specify and still fail in practice. We must cover this subject one way or another, it seems to me, either by letting it stand or by requiring that if the manufacturer departs from our specification, he state just what his departure is.

MR. MANLY:—Cannot these tests be so changed as to insure that there shall be no substances used deleterious to the life of the hose?

MR. HALE:—I think that tests do not protect the goods. The thought is to embody, first, the fact that the rubber shall be hevea, wild or plantation; second, that there shall not be less than 25 per cent of rubber (rubber adhesive between fabrics, either in the tube, friction, or cover compounds); and, third, the amount of free sulphur shall not exceed 1 per cent.

(On motion, duly seconded, it was voted that the paragraph on "Chemical Properties" be revised as suggested by Mr. Hale.)

H. M. CRANE (M. S. A. E.):—The second paragraph under the heading "Inner Tube," states: "When of an internal diameter of ¾ in. or less, this inner tube must be seamless. Tubes of larger internal diameter may be made from rubber sheeting, in which case they must consist of at least two complete turns." Tubes made up in that way, not cemented or vulcanized down would probably curl and close up if the weather were to swell them at all. It is not clear to me whether that specification refers to uncemented or unvulcanized tubes. I think it is a great mistake to leave the rubber loose.

MR. TUTTLE:—Nobody thought that a hose could be made any other way than by vulcanizing it into one

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complete tube. When we apply permeability tests, we shall find out what the term "vulcanized rubber" means. There is no doubt about it in the manufacturer's mind.

MR. HALE:—In manufacturing large-size hose it is customary to have the green stock rubber in a sheet; this is rolled in two turns on a pole. The fabric is then wound around and the tube put outside of it. The rubber is wrapped in the green stage and then vulcanized. The hose will be just as uniform and homogeneous as though made in the form of a tube in the first place.

MR. CRANE:—It seemed to me that these specifications are composed of terms understood in the trade. If we allow them to contain too much of what is expected to become practice, there will be an opportunity for inspectors to cause trouble, because usually they do not know anything about trade practice.

W. S. WOLFE (M. S. A. E.):—Does not the fact that the inner tube is to be made of rubber free from flaws and cracks cover the matter? I move that the sentence in question read as follows: "Tubes of larger internal diameter may be made of rubber sheeting, in which case they must consist of at least two turns and be formed into a complete tube in the further processes of manufacture."

(The amendment to apply to both specifications was carried)

Nuts for Aircraft

CHAIRMAN UTZ:—I will ask Mr. Ehrman to discuss the subject of nuts. In the Division itself we thought that the chamfer on the bottom of all of them was a good thing.

E. H. EHRMAN (M. S. A. E.):—I have checked up that detail, and find that the length of the hexagon nut with bottom chamfer is inside of our safety point which is the diameter of the nut divided by four.

The thickness of the nut, which represents the length of thread bearing, has been made such that it is three-quarters the normal size of the bolt, in all instances comparing with the present S. A. E. castle nut with the castellated part removed, and in the smaller size, with the specifications now under consideration. In all of these specifications, the nominal size of the bolt should be expressed in four decimal places, inasmuch as the interchangeability depends upon having a uniform dimension from which to work. The dimensions expressed in three decimal places are often fifteen thousandths of an inch

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large, and if followed it would be impossible to have a basis for any standard.

MR. MANLY:—I think Mr. Ehrman's suggestion in regard to the necessity for expressing the four decimal places is well made. The basic size of the bolt itself should exceed the nominal size. It will cause serious interference if it is not strictly adhered to in connection with the screw thread.

MR. EHRMAN:—I move to have all of the size A dimensions of nuts and bolts that are given in decimals expressed to the fourth decimal place and that wherever the decimals have equivalent common fractions the latter be shown in parenthesis.

(The motion proposed by Mr. Ehrman was seconded and carried)

Spark-Plug Shell

O. J. ROHDE (M. S. A. E.):—Some manufacturers are using gages to measure the threads of the shell while others are using the thread micrometers. The tap and die manufacturers are using one system of checking, the airplane people are using a different system, and they do not agree on the finished product.

MR. MANLY:—I think it is unnecessary for the Society to tell people how to measure threads. The whole thing depends on whether the gages are really standard. The length of the threaded portion is the chief question. About as many different lengths have been wanted as there are designers of engines, and the only thing that we have been able to agree upon at all has been the length specified here, which is a compromise. We tried to select what seems to be a usable length for proper results, and it is hoped that designers will commence to use this as a standard. I believe that this $\frac{5}{8}$-in. length meets all requirements and also carries away the heat. I have seen some engines work very hot under high compression, and, so far as I know, this length has been sufficient to carry off the heat in case it goes to the plugs.

MR. ROHDE:—In automobile practice the standard has been set now for a few years at $\frac{5}{8}$ in. maximum length and automobile engineers have gradually swung over to that standard. Whenever longer plugs are required, they are made as special plugs and are put on the market as such. The ability to conduct heat is more a question of actual insulating efficiency than of thread length.

MR. MANLY:—It is a question of making the threads themselves accurate so that there will be an actual contact between the plug thread and the thread in the cylinder.

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If the thread has a proper slope in both cases the ⅝-in. length will result in the heat being carried away. I have not heard of any trouble on that score. There will be as much contact with a loose thread as with a tight thread, if the thread in the cylinder and the thread on the plug have the same angle of slope.

MR. EHRMAN:—The chamfer at one end of the thread is 30 and at the other 35 degrees. At an earlier meeting 35 deg. was adopted for the chamfer at the lower end and presumably for that at the upper end. If some manufacturers are using 35 and others 40 deg., why not standardize on the former, so as to have a consistent design, the angles at both ends being the same.

(On motion, duly seconded, it was voted to recommend that the angle of chamfer be 35 deg. at each end of the thread)

Bevel Washers

MR. MANLY:—The Subdivision found that various bevels are being used, and that bevel washers are expensive to manufacture because of the milling machine operations required. We have, therefore, tried to work out something that would lend itself a little better to manufacturing.

Two types of bevel washers are presented, one the regular round and the other a square washer. The value of a square washer is that by using the four angles, 6, 12, 24 and 36 deg., we obtain bevels to meet practically all known conditions, and certainly can secure this result by combining two of the washers. In order to facilitate manufacture, since on the high angles it is impossible to stamp out material to the round, a square washer is suggested. This would be made from rolled bevel stock by merely cutting it off with a gang-saw. It has other very valuable qualities. One advantage is that the washer can be set quickly, so that the line of the bevel is placed in the direction desired; another is that in tightening down the nut the washer is easily held and kept from twisting around.

For that reason both the square and round washers are recommended, and the designer can determine which he prefers to use.

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FIRST REPORT OF MOTORCYCLE DIVISION

(*As Adopted by the Society*)

THE following list of standards is recommended by the Division, to be incorporated in the Government military motorcycle specifications for all service:

Spark control on military motorcycles shall be of the handlebar-grip type placed on the left-hand side and so arranged that the spark will be advanced when the top of the grip is turned toward the center.

Throttle control on military motorcycles shall be of the handlebar-grip type, placed on the right-hand side and arranged so that the throttle will be opened when the top of the grip is twisted toward the center.

Kick starters on military motorcycles shall be of the folding-pedal type, so arranged as to operate when pushed down and back.

(It is believed by the Division that starters cannot be confined to a definite side until further standardization of engines takes place)

Clutch pedals on military motorcycles shall be placed on the left-hand side and shall act to release the clutch when the pedal is pushed forward and down, clutching by reverse action.

Brake pedals on military motorcycles shall be placed on the right-hand side so that the brake will be applied when it is pushed down.

Driving Chains on military motorcycles shall be of the roller type, with 5/8-in. pitch; roller width 3/8 in.; and roller diameter, 0.40 in.

Oil and grease cups on military motorcycles shall have an attaching diameter of 5/16 inch with 32 threads.

Military motorcycles shall have an engine displacement of not less than 60 nor more than 61 cubic inches.

Gearshifts on military motorcycles shall be placed on the left-hand side and be of the progressive type.

(As several methods are now in use for operating gearshifts, namely by movement of the lever up and down, or forward and backward, with high-gear position placed either front or rear, the Division considers it advisable that further standardization be held in abeyance for the time being)

Carrying Capacity The maximum sprung load-carrying capacity on military motorcycles and side-cars shall not exceed 500 lb., this including the operator; no load to

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be strapped or attached to any part of the frame of either the motorcycle or side-car, except that two rifles and boots may be attached to the front fork.

Tires on military motorcycles, rear and side-cars, shall be of the clincher type, 28 by 3 inch.

Magneto Dimensions In arriving at the magneto dimensions recommended below the leading makes of magnetos were studied and the space allowed on the motorcycles given careful consideration.

	Mm.	In.
Shaft height	44.98	1.771
Distance from center of front base-plate holes to large end of shaft taper	39.45	1.553
Distance from center of front base-plate holes to center of rear base-plate holes	50.0	1.968
Distance between centers of base-plate holes, left to right	50.0	1.968
Large diameter of taper	13.0	0.5118
Small diameter of taper	11.2	0.4409
Length of taper	9.0	0.3543
Taper 1:5 (included angle) 11 deg. 30 min. approx. Woodruff key special 5/16 by 5/32 by 3/32-in. thick, set to project 1/16 in.		
Height of magneto space	152.4	6.000
Length of magneto space (from large end of shaft taper)	161.93	6.375
Width of magneto space	95.25	3.750
Width at brushes	101.60	4.000
Diameter plain hole timing lever	5.55	0.2185
Base-plate holes 3/8-in., 16 threads per inch, U. S. Standard.		
Thread for end of magneto shaft 5/16 in. dia., 18 threads per inch U. S. form. Length of thread, 0.417 in.		
Advance lever, radius	50.0	1.968

Spoke and Nipple The general dimensions and tolerances of the finished spokes and nipples for military motorcycle wheels shall be as shown in the accompanying drawing. The thread tolerance shown is without plating.

Threads may be either cut or rolled, but the unthreaded portion of butt must remain 0.148, + 0.003, — 0.000 inch.

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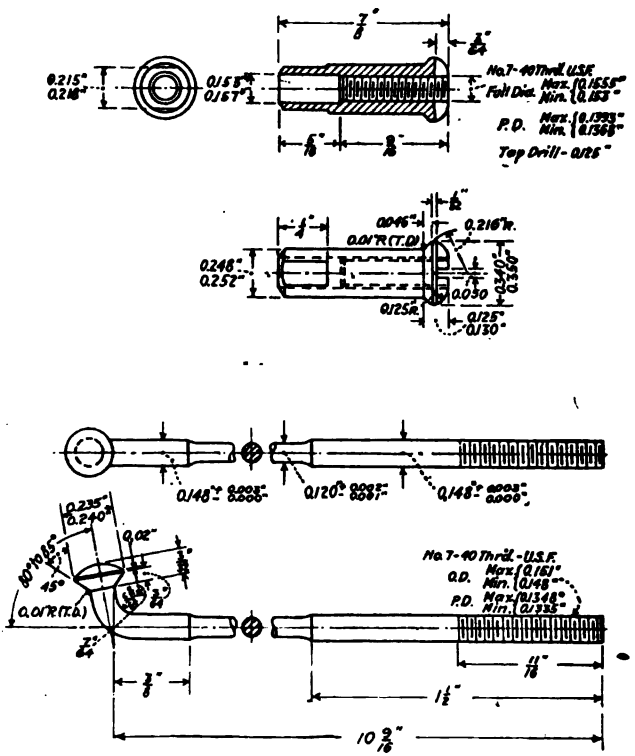

SPOKE AND NIPPLE DIMENSIONS

The nipple is to be made from free-cutting steel screw stock and is to be finished according to dimensions.

Spoke-Wire Specification The spoke wire shall have the following chemical content within the limits specified:

Carbon, 0.40 to 0.55; manganese, not under, 0.50; phosphorus, not over, 0.05; sulphur, not over, 0.05.

Tolerance of Diameter The nominal wire diameter must be 0.148, + 0.000, — 0.001 inch.

Bend Test The wire must be held firmly in a pair of jaws so that it may be bent over a surface having a radius of one-half the nominal diameter of the wire, and shall stand a test of at least four bends, counted as follows: Holding the free length of the wire loosely between clamps so that it cannot bend except at the holding jaws, it is to be bent 90 deg. over one jaw; this to be counted as one bend. It is then to be bent up to its original position; this to be counted as the second bend. It is then

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to be bent 90 deg. to the other side (third bend) and back to its original position to complete the test.

Tensile Strength The tensile strength of the wire is to be not less than 140,000 lb. per sq. in.

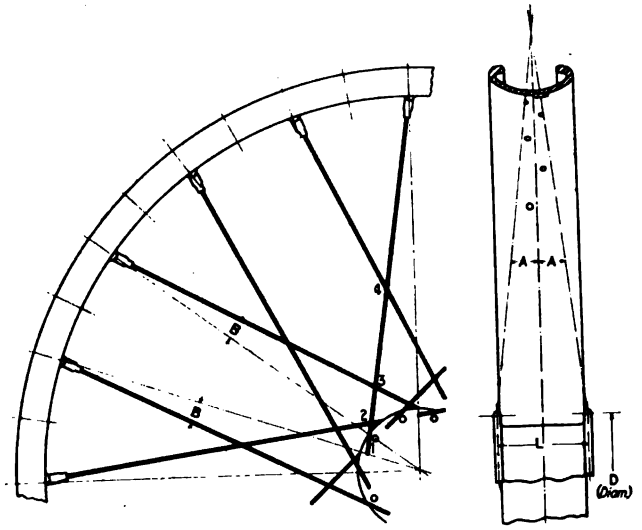
Tensile Strength of Complete Spoke and Nipple The finished spoke, when held at the bent end in the same manner as by the hub flange in a finished wheel and at the threaded end by a standard nipple with the spoke threads filling all the nipple threads, must sustain a dead load of not less than 1580 lb. No torsion test is required.

Wheel and Rim Specification This provides for a 28 by 3-in. clincher tire with 40 spokes laced four across. The circumference of the wire-wheel rim after lacing must be 68.722, + 0.047 (3/64), — 0.172 (9/64) in.

The chemical content of the steel used for the rims must be:

Carbon, 0.10 to 0.15; sulphur, not over, 0.05; phosphorus, not over, 0.05; manganese, 0.40 to 0.60.

Fuel and Lubrication Pipe Fittings Fuel and lubrica-



DIMENSIONS FOR MOTORCYCLE-WHEEL LACING AND HUBS

	A Deg. Min.	*B* Deg. Min.	*D* In.	*L* In.	*I* In.	*K* In.
Nominal...	6°	7° 30′	2.86	2.93	0.165	0.255
Maximum..	7° 45′	9° 15′	3.56	3.60	0.180	0.263
Minimum..	4° 15′	5° 45′	2.20	2.25	0.150	0.247

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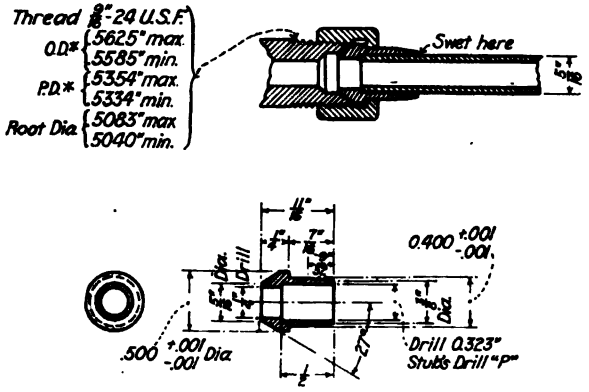
tion pipes on military motorcycles shall be 5/16 in. outside diameter. The fittings shall be of the soldered type, with 9/16-in. nut, 24 threads to the inch, United States form.

Headlamp Mounting The mounting lugs and supporting prongs shown in the accompanying sketch are recommended; these conform to existing practice.



..*Actual size
†Minimum diameter for respective pitch diameters
Outside diameter tolerances of commercial taps are slightly greater

FUEL AND LUBRICATION PIPE FITTING DIMENSIONS

DISCUSSION AT STANDARDS COMMITTEE MEETING

W. S. HARLEY (M. S. A. E.): The work of the Motorcycle Division was brought about at the request of the Quartermaster Department, and applies essentially to military motorcycles. Up to the time of this request,

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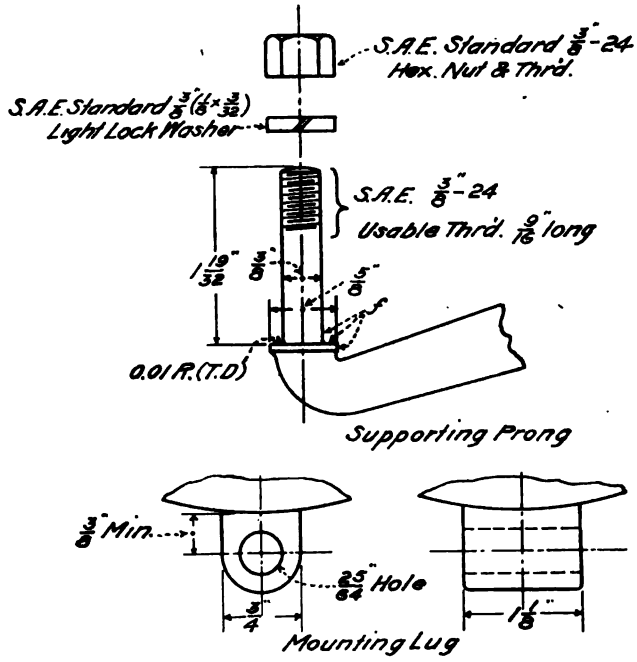

HEADLAMP LUGS AND SUPPORTING-PRONG DIMENSIONS

there were no standards in the motorcycle industry, but the Government is using two or three different makes of motorcycles, so that it was felt there should be as much standardization on these as possible.

This report covers primarily motorcycles used by the Government and is an endeavor to standardize on all the parts possible in those motorcycles without requiring radical changes in design.

JEROME J. AULL (M. S. A. E.): The threatened multiplication of union standards is serious. First, we had the S. A. E. standard, now we have the motorcycle standard. The Tractor Division has another, and the Aeronautic Division will have another. The Signal Corps has adopted a union similar to the one proposed for motorcycles. In this particular case, the thread is ½ in. by 24 to the inch. The Signal Corps union will be ½ in. by 20 threads to the inch, which is the S. A. E. thread. We ought to bring these Divisions together in some way.

MR. HARLEY: In laying out this union, we tried to adopt that standard ½ in. "twenty," but found it to be too coarse. There are three pipes on the motorcycle, a

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pipe from the hand pump, one from the mechanical oil pump and a gasoline pipe. All those pipes have been different, and we adopted one pipe for all three purposes, with a view to simplification. When using a "twenty" thread, the nut, as it is now, is too big for motorcycle work. It does not look well; furthermore, in automatic work, a 24-thread is much better than a 20. From a shop point of view the work is easier to tap. Then too, generally speaking, the motorcycle cannot use S. A. E. standards, especially over $\frac{5}{8}$ in., because the threads are too coarse. Because of the vibration we have to use threads finer than the S. A. E. standard.

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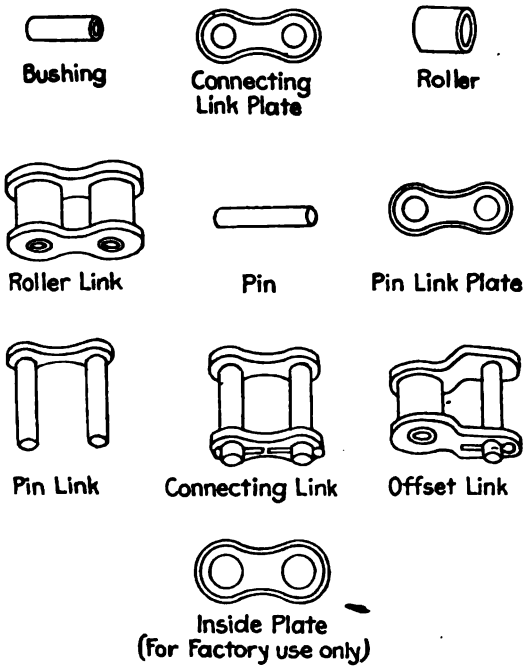
FIRST REPORT OF ROLLER-CHAIN DIVISION

(As Adopted by the Society)

The nomenclature for component chain parts, as shown in the accompanying drawing, is recommended in order to establish universal practice among both the manufacturers and the trade. The Division has worked jointly



NOMENCLATURE FOR ROLLER CHAIN PARTS

with the A. S. M. E. Chain Committee in making these recommendations, and also in preparing the dimensions included in this report.

Roller Chain Dimensions The sizes recommended are selected from those most generally in use, with the addition of a 2½-in. pitch for heavier duty. Other sizes will be worked out for light duty at high speeds, and heavy duty at slow speeds, and will be influenced more or less by the future introduction of alloy steels in chain manu-

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facture. The chain-size numbers indicate the number of quarter inches in the pitch, and the word "Universal" indicates recommended standard series.

TABLE OF ROLLER CHAIN DIMENSIONS

Trade No.	Pitch	Chain Width*	Roller Diameter
Universal 2	½	¼	0.306
Universal 2½	⅝	⅜	0.400
Universal 3	¾	½	15/32
Universal 4	1	½	⅝
Universal 5	1¼	⅝	¾
Universal 6	1½	1	⅞
Universal 7	1¾	1	1
Universal 8	2	1¼	1¼
Universal 10	2½	1½	1 9/16

*Distance between the inside plates All dimensions are in inches

DISCUSSION AT STANDARDS COMMITTEE MEETING

WILLIAM F. COLE (M. S. A. E.): For a long time there has been no uniformity in the nomenclature applied to the parts of the roller chain. This condition has frequently resulted in misunderstanding and confusion in filling orders and has led to the suggestion of having a standard nomenclature for these parts.

CHAIRMAN UTZ: The specifications cover general standard practice, do they not?

MR. COLE: Yes. They represent the chains that are most in use at the present time. It was the unanimous belief of the trade members that at this time, when tool makers are employed on important work, it would be unwise to introduce a new set of standard sizes, which would call for the making of new tools and dies.

We found that probably five times as many sizes of roller chains are in general use as are required, because each manufacturer has duplicated sizes originally made by others. This was due to a natural tendency to shorten the pitch of the chains and make them take care of higher speeds.

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SECOND REPORT OF TRACTOR DIVISION

(As Adopted by the Society)

The present report includes recommendations on drawbar connectors, magneto mounting, use of S. A. E. standards in tractor service, and a condensed tractor specification form.

Magneto Mounting It is recommended that the National Gas Engine Association standard distance of 2.375 in. from the large end of the magneto shaft taper to a point ¾ in. back from the end of the ¾-in. engine shaft be standardized. When magneto is to be mounted on an iron or steel bracket a shim at least ½ in. thick, made from brass, aluminum, or other non-magnetic material, should be used and be furnished by the engine builder, so that the standard distance from top of shim to center of magneto shaft will be 1.771 in. (the height previously standardized by the Society), and also that brass bolts be used to hold down the magneto.

S. A. E. Standards for Tractor Service The Division recommends the following list of existing S. A. E. standards (Vol. I, S. A. E. Handbook) as suitable for tractor work:

Name	Data Sheet No.
Spark-plug shells	3
Screws and bolts ¼ to 1½ in.	4
Sizes of taps and drills	4a
Large diameter thread (1⅝ to 6 in.)	4c
Lock washers	5
Square broached fittings	7
Taper fittings	7a
Taper fittings with castle nuts	7e
Steels and heat treatments	9 to 9q
Babbitt metal	11
Brass casting metals	12
Cast manganese bronze	12a
Manganese bronze sheets and rods	12b, c
Hard cast and gear bronze	12d
Aluminum alloys	13
Non-ferrous metal tubing	13c
Round tension test specimen	15
Flat tension test specimen	15xa
Flat tension test specimen	15a
Shock test specimen	15xb
Gray-iron test specimen	15b

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Gray-iron test specimen.....15c
Brinell hardness test.....15xd, d
Cold-drawn seamless steel tubes.....16
Bands and strips.....17
Ball and roller bearings.....29 to 29e
Clamps and fittings for rubber hose.....37
Oversize cylinders.....47

Tractor Specification The following condensed specification is recommended for standardization:

1 Firm name.....
Address .....
2 Tractor trade name.....model.....
3 Drawbar horsepower S. A. E. tractor rating.....
Old trade rating.....
4 Belt horsepower S. A. E. tractor rating.....
Old trade rating.....
5 Engine: Make.....in. bore.....stroke.....in.
No. cylinders.....cycle.....normal r.p.m.....
Lubricating system.....type.....
Ignition .....Carbureter.....
Fuel system.....
Cooling system.....
Belt pulley, dia.....in.....face.....in.....rpm.
6 Transmission type or make.....
No. speeds forward.....
Speed in m.p.h. at normal engine speed—
1st .....2d.....3d.....
Indicate by check (x) speed normally used for plowing
7 Wheels: No.....arrangement.....
Driving No.....dia.....in..face.....in.
Non-driving No.....dia.....in.....face.....in.
If of track-laying type, No. of trucks.....
Length.....in. Face.....in.
8 Frame construction.....
9 Wheelbase .....in....gage.....in.
10 Width overall.....in. Length overall.....in.
11 Turning radius (½ dia. largest track circle).....ft.
12 Wight of tractor, les fuel, oil, water and lugs....lb.
13 Shipping weight, including standard equipment and stays .....lb.

Tractor and Implement Drawbar Connectors It is recommended that for tractors with 20 drawbar hp. or less the tractor drawbar be horizontal and ⅝ in. thick, with 13/16-in. holes located with centers 1 in. from the

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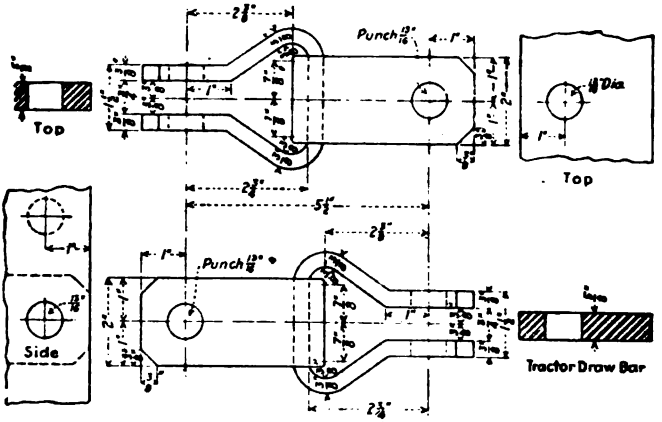
edge of the bar; that implements be made with drawbars vertical, 5/8 in. thick and with 13/16-in. holes located with centers 1 in. from the edge of the bar; and that a number of additional holes be provided in both tractor



TRACTOR AND IMPLEMENT DRAWBAR CONNECTOR DIMENSIONS

and implement drawbars to take care of any adjustment found necessary.

DISCUSSION AT STANDARDS COMMITTEE MEETING

DENT PARRETT (M. S. A. E.): The dimensions of the drawbar design are not recommended as standard; they are simply shown as a possible way to make a hitch, or to connect the standard drawbar on the implement and on the tractor. The real point of these connections is to encourage the use of a horizontal drawbar on the tractor.

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SECOND REPORT OF DATA SHEET DIVISION

(As Adopted by the Society)

The Data Sheet Division submits recommendations on a number of minor changes in the S. A. E. Handbook.

Cancelled Data Sheets It is recommended that Data Sheets 6g (Finish of Hexagon Cap Screws) and 51 (French Road Signals), Vol. I, S. A. E. Handbook, be cancelled.

Changes in Nomenclature (a) It is recommended that on Data Sheet 8b, Vol. I, the heading "Industrial Truck Wheels" be changed to be "Industrial Truck Tires," and that the heading "Wheel Dimensions" in the first column of the table be changed to "Tire Dimensions." (b) It is recommended that wherever the expression "pleasure car" occurs in the Data Sheets the expression "passenger car" be substituted.

Topical Index An up-to-date general index has been completed for the Data Sheets and mailed to the members of the Society, together with the latest Data Sheets.

The Division recommends that separate index sheets or sections of the general index include the general Standards and Recommended Practices which may be approved for use in each of the following subjects: Aeronautic, Marine, Motorcycle, Stationary and Farm Engine, and Tractor.

Data Sheet Size It has been suggested that the Data Sheets be enlarged from their present size, but after careful consideration the Division unanimously voted to retain the present size, for the time being at least.

ELEVENTH REPORT OF MISCELLANEOUS DIVISION

(As Adopted by the Society)

Screw Threads The fine pitch threads for screws from ¼ to 1½ in., shown in the accompanying table, are submitted for approval as standard practice. These, taken with the threads given on Data Sheets 4 and 4c, Vol. I, S. A. E. Handbook, complete the list of S. A. E. screw standards. All these threads are to be of the U. S. Standard form.

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TABLE OF SCREW THREAD DIMENSIONS

Diameter, In.	Regular Pitch	Fine Pitch	Diameter, In.	Regular Pitch	Fine Pitch
¼	28	36	3	10	16
9/32	24	32	3¼	10	16
⅜	24	32	3½	10	16
7/16	20	28	3¾	10	16
½	20	28	3½	10	16
9/16	18	24	3¾	10	16
⅝	18	24	3¾	10	16
11/16	16	24	3¾	10	16
¾	16	20	4	10	16
⅞	14	20	4¼	10	16
1	14	20	4¼	10	16
1⅛	12	18	4½	10	16
1¼	12	18	4½	10	16
1⅜	12	18	4¾	10	16
1½	12	18	4¾	10	16
1⅝	12	16	4⅞	10	16
1¾	12	16	5	10	16
1⅞	12	16	5¼	10	16
2	12	16	5¼	10	16
2⅛	12	16	5½	10	16
2¼	12	16	5½	10	16
2⅜	12	16	5¾	10	16
2½	12	16	5¾	10	16
2⅝	12	16	6 & over	8	16
2¾	12	16			

NOTE—The maximum screw size equals nominal or basic screw size (for all except wrench fits). In the minimum gage for internal threads and the maximum gage for external threads, the profile of the threads shall be such as not to encroach on that of the true United States Standard (or Franklin Institute) thread. These recommendations are made to insure interchangeability and do not pertain to dimensions or clearances.

Bumpers and Mounting After careful review of the proper height of both front and rear bumper for various conditions of loading, fender clearance and collision, the Division recommends the following table of heights and dimensions for standardization:

BUMPER HEIGHTS AND DIMENSIONS

	Front	Rear
Height, center of face to ground	21	22
Overall length	Between 59 and 60	Between 59 and 60
Bumper face	Flat	Flat
Vertical depth of bumper bar (min.)	2	2½

All dimensions in inches

Yoke and Eye Rod-Ends The dimensions* for adjustable yoke rod-ends, plain yoke rod-ends and eye rod-ends, previously listed as "future sizes," are now submitted for approval as S. A. E. Standards.

*See report of Miscellaneous Division, S. A. E. TRANSACTIONS, 1917, Part I, and S. A. E. Handbook, Vol. I, Sh. 1, 1a and 2

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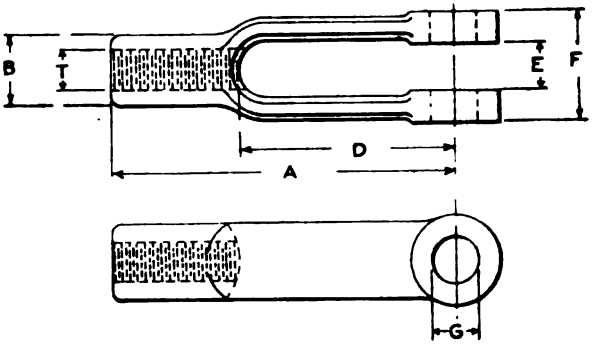
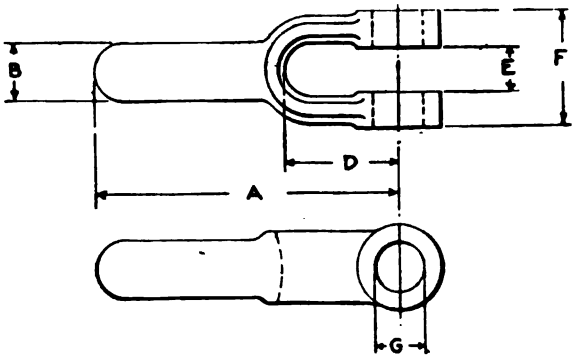


ADJUSTABLE YOKE ROD-END DIMENSIONS

T	A	B	D	E	F	G	Limits o
½—14	7½	1 5/16	5¼	1 9/16	1⅝	½	+0.001 −0.002
1—14	8	1½	6	1 13/16	2⅛	1	+0.001 −0.002

Threads U. S. Form



PLAIN YOKE ROD-END DIMENSIONS

B	A	D	E	F	G	Limits on G
½	4	2	1 9/16	1⅝	½	+0.001 −0.002
1	4½	2½	1 13/16	2⅛	1	+0.001 −0.002

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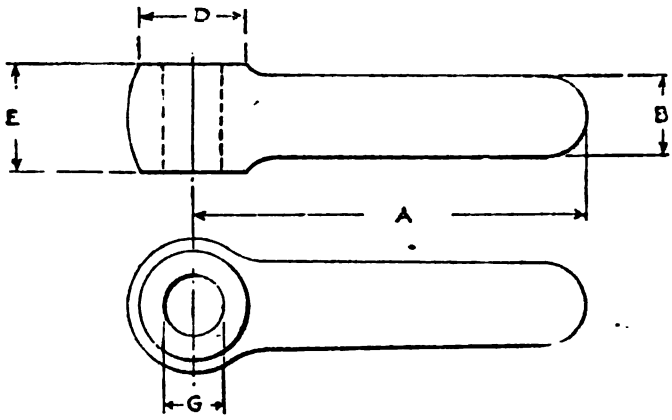
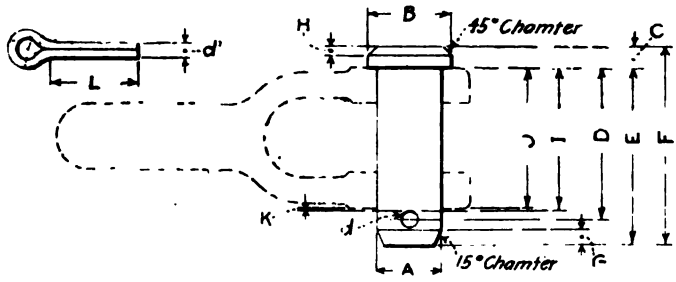


EYE ROD-END DIMENSIONS

A	B	D	E	G	Limits of G
2⅛	⅞	1 1⅛	1 1⅛	⅞	+0.001 −0.002
3	1	1 1⅛	1 1⅛	1	+0.001 −0.002

Rod-End Pins The following extension of rod-end pin dimensions* is recommended. This extension is for the pins required for the yoke and eye rod-ends, sizes ⅞ and 1 inch.



*See report of Miscellaneous Division, S. A. E. TRANSACTIONS, 1917, Part II. p 44, and S. A. E. Handbook, Vol. I, sh. 2a

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TABLE OF ROD-END PIN DIMENSIONS

A	Limits of A	B	C	D		E	F	G	H
⅞	−0.004 −0.007	1 5/16	5/16	2 1/16	2.0156	2 1 1/32	2 1 1/32	5/16	3/32
1	−0.004 −0.007	1 5/16	1 1/32	2 1 3/16	2.2563	2 1/4	2 1 1/32	3/32	3/64

A	I	S. A. E. Standard Yoke J	K	Cotter Pin		d = Drill		Difference	
				d'	L	No.	Dia.	d − d'	*d − d' + b
⅞	1.9201	1 15/16	0.0451	1 1/64	1 1/2	11	0.191	0.0191	0.0642
1	2.1701	2 1/16	0.0451	1 1/64	1 1/2	11	0.191	0.0191	0.0642

*Total clearance between yoke and cotter-pin

SECOND REPORT OF MARINE DIVISION

(As Adopted by the Society)

The report of the Marine Division contains recommendations relating to flange steel couplings (between engine and reverse gear, and reverse gear and propeller-shaft), clamp steel coupling (propeller-shaft extension), flange cast iron coupling (reverse gear and propeller-shaft), and propeller mountings.

In the accompanying drawing the engine flange when integral with the engine shaft is shown in full lines. When separate, the shaft taper and dimensions are the same as for reverse gear coupling of the corresponding size shown in Table II.

The reverse gear flange shown by full lines shall have drilled bolt holes. When the flange is designed integral with the reverse gear, as shown by dotted lines, all bolt holes shall be drilled and tapped for S. A. E. standard threads, as shown on Data Sheet 4, Vol. I, S. A. E. Handbook.

These couplings are designed to be made of steel and are intended for high power engines.

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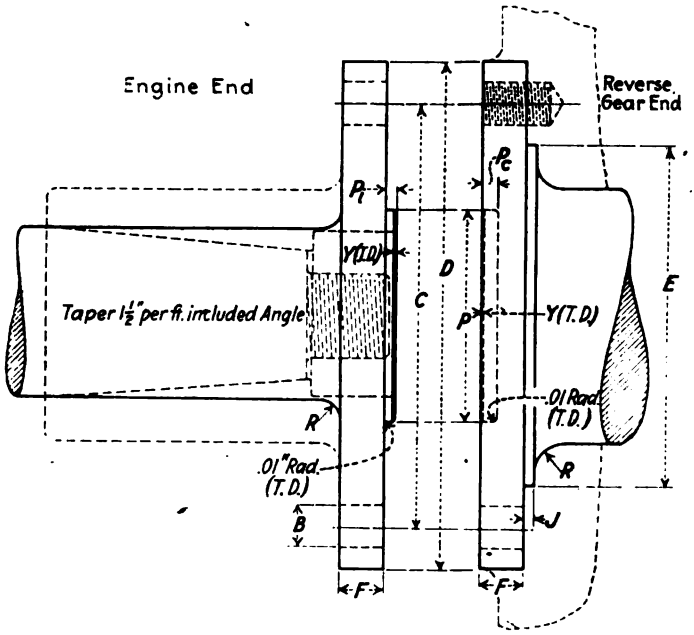
FLANGE STEEL COUPLING

(Reverse Gear and Propeller-Shaft)

All bolts shall have S. A. E. standard threads as shown on Data Sheet 4, Vol. I, S. A. E. Handbook. (Safety sleeves or covers for these couplings are being considered by the Marine Division)

FLANGE STEEL COUPLING

(Engine and Reverse Gear)



SHAFT (CLAMP) COUPLING

(Propeller-shaft extension)

The clamp coupling shown in the accompanying drawing is for propeller-shaft extension and is not intended for coupling propeller-shaft directly to the reverse gear. Unless otherwise specified S. A. E. manganese bronze No. 29 should be used. All bolts shall have S. A. E. standard threads as shown on Data Sheet 4, Vol. I, S. A. E. Handbook. (Safety sleeves or covers for these couplings are being considered by the Division)

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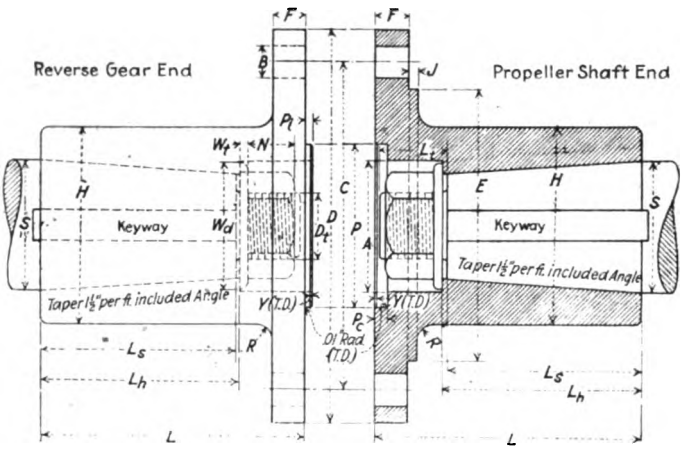

STEEL REVERSE COUPLING

TABLE I—FLANGE STEEL COUPLING (ENGINE AND REVERSE GEAR)

Size No. See Note	Equiv. Shaft Dia.	Flange						Pilot *				Bolts		
		D	*E*	*F*	*J*	*R*	*Pc*	*Pc*	*Pc*	*Y*	No.	*B*†	Thds.‡ per Inch	*C*
ER- 8	1	4	2 1/32	3/8	1/4	3/16	1 5/8	1/16	1/8	1/64	4	5/8	24	3
ER-9	1 1/8													
ER-10	1 1/4	5	3 1/32	7/16	3/16	5/16	2	1/8	5/16	1/32	6	7/16	20	4
ER-11	1 3/8													
ER-12	1 1/2													
ER-13	1 5/8													
ER-14	1 3/4	6	4 1/16	1/2	1/4	1/4	2 1/2	1/8	3/16	1/32	6	9/16	18	5
ER-15	1 7/8													
ER-16	2													
ER-18	2 1/4	7	4 1/2	5/8	5/16	5/16	3	5/16	1/4	1/32	6	5/8	18	5 1/2
ER-20	2 1/2													
ER-22	2 3/4	8	5 5/8	3/4	3/8	3/8	3 3/8	5/16	1/4	1/32	6	3/4	16	6 1/2
ER-24	3													
ER-26	3 1/4	9	6 1/16	7/8	3/8	7/16	4 3/8	1/4	5/16	3/64	6	7/8	14	7 1/4
ER-28	3 1/2													
ER-32	4	10	6 1/2	1	1/2	1/2	5	5/16	3/8	3/64	6	1	14	8

*Pilot tolerance +.000, −.001 in. per inch of diameter. Pilot-recess tolerance, +.001; −.000 in. per inch of diameter

†Figures in column indicate bolt diameters. Use clearance drill of same nominal size for holes *B*

‡All threads S. A. E. pitch, U. S. form. (See S. A. E. Handbook, Vol. I, sh. 4)

(T. D.) = Tooling dimensions. All dimensions in inches

NOTE.—E indicates engine; R, reverse gear; and figures, number of eighths of an inch in equivalent shaft diameters

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TABLE II—DIMENSIONS OF FLANGE STEEL COUPLING (REVERSE GEAR AND PROPELLER-SHAFT)

Size No.	SHAFT				NUT‡		KEYWAY		WASHER		FLANGE					PILOT				HUB				BOLTS		
	S	*Ls*	*Dt*	*Lt*	*N*	Thds. per. in.	Width	Height	*Wt*	*Wd*	*D*	*E*	*F*	*J*	*A*	*P**	*Pl*	*Pc*	*Y*	*H*	*L*	*Lh*	*R*	No.	*B*†	*C*
RPS- 8	1	1 1/16	9/16	1/2	7/16	18	1/4	1/8	1 3/16	1/8	4	2 1/32	3/8	1/16	1 3/8	1 5/8	1/16	1/8	1/64	1 7/8	2 1/2	1 7/8	3/16	4	3/8	3
RPS- 9	1 1/8	1 1/16	9/16	1/2	7/16	18	1/4	1/8	1 5/16	1/8	4	2 1/32	3/8	1/16	1 3/8	1 5/8	1/16	1/8	1/64	1 7/8	2 1/2	1 7/8	3/16	4	3/8	3
RPS-10	1 1/4	2 1/16	5/8	5/8	9/16	16	5/16	5/32	1 7/16	1/8	5	3 1/32	7/16	5/32	1 3/4	2	1/8	3/16	1/32	2 3/4	3	2 1/4	1/32	6	5/16	4
RPS-11	1 3/8	2 1/16	5/8	5/8	9/16	16	3/8	3/16	1 7/16	1/8	5	3 1/32	7/16	5/32	1 3/4	2	1/8	3/16	1/32	2 3/4	3	2 1/4	1/32	6	5/16	4
RPS-12	1 1/2	2 1/16	5/8	5/8	9/16	16	3/8	3/16	1 7/16	1/8	5	3 1/32	7/16	5/32	1 3/4	2	1/8	3/16	1/32	2 3/4	3	2 1/4	1/32	6	5/16	4
RPS-13	1 5/8	2 1/16	5/8	5/8	9/16	16	3/8	3/16	1 7/16	1/8	5	3 1/32	7/16	5/32	1 3/4	2	1/8	3/16	1/32	2 3/4	3	2 1/4	1/32	6	5/16	4
RPS-14	1 3/4	2 1/16	1	1	3/4	14	7/16	3/32	1 1/16	1/8	6	4 1/16	1/2	1/8	2 1/16	2 1/2	1/8	3/16	1/32	3	4	3	1/4	6	5/16	5
RPS-15	1 7/8	2 1/16	1	1	3/4	14	1/2	1/4	1 1/16	1/8	6	4 1/16	1/2	1/8	2 1/16	2 1/2	1/8	3/16	1/32	3	4	3	1/4	6	5/16	5
RPS-16	2	2 1/16	1	1	3/4	14	1/2	1/4	1 1/16	1/16	6	4 1/16	1/2	1/8	2 1/16	2 1/2	1/8	3/16	1/32	3	4	3	1/4	6	5/16	5
RPS-18	2 1/4	3 1/16	1 1/8	1 1/8	1 1/16	12	5/16	5/16	2 5/16	3/16	7	4 1/2	5/8	1/8	2 1/2	3	1/16	1/4	1/32	3 3/4	5	3 7/8	1/16	6	1/2	5 1/2
RPS-20	2 1/2	3 1/16	1 1/8	1 1/8	1 5/16	12	1/2	1/16	2 5/16	3/16	7	4 1/2	5/8	1/8	2 1/2	3	1/16	1/4	1/32	3 3/4	5	3 7/8	1/16	6	1/2	5 1/2
RPS-22	2 3/4	4 1/16	1 3/8	1 1/4	1/8	12	3/4	3/8	2 1/16	1/4	8	5 3/8	3/4	1/8	3	3 3/4	1/4	5/16	3/64	4 3/8	5 1/2	4 1/4	3/8	6	3/4	6 1/2
RPS-24	3	4 3/8	1 3/8	1 1/4	1/8	12	3/4	3/8	2 1/16	1/4	8	5 3/8	3/4	1/8	3	3 3/4	1/4	5/16	3/64	4 3/8	6	4 3/4	3/8	6	3/4	6 1/2
RPS-26	3 1/4	5 3/16	1 1/2	1 3/8	1/8	12	3/8	3/16	3 3/16	1/4	9	6 1/16	1/8	3/16	3 3/4	4 3/8	1/4	5/16	3/64	5	7	5 3/8	3/16	6	7/8	7 3/8
RPS-28	3 1/2	5 3/16	1 3/4	1 3/8	1/8	12	7/16	3/16	3 3/16	1/4	9	6 1/16	1/8	3/16	3 3/4	4 3/8	1/4	5/16	3/64	5	7	5 3/8	3/16	6	7/8	7 3/8
RPS-32	4	6 1/4	2	1 1/2	1 5/16	12	1	1/2	3 1/16	5/16	10	6 1/2	1	3/16	4 1/4	5	5/16	1/8	3/64	6	8	6 1/2	1/2	6	1	8

*Pilot tolerance, +0.000, −0.001 in. per inch of diameter. Pilot-recess tolerance, +0.001, −0.000 in. per inch of diameter
†Figures in column indicate bolt diameters. Use clearance drill of same nominal size for holes *B*
‡Nuts for Style Nos. RPC 26 and RPC 28 use 2⅛ in. hex stock. Nuts for Style No. RPC 32 use 3 in. hex stock
(T. D.) = Tooling dimensions. All dimensions in inches
NOTE—R indicates reverse gear; P, propeller-shaft; S, steel; and figures, number of eighths of an inch in shaft diameter

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TABLE III—DIMENSIONS OF SHAFT (CLAMP) COUPLINGS (PROPELLER-SHAFT EXTENSION)

Size No. (See Note)	S	KEYWAY		CLAMP								BOLTS					
		W	h	L	D	E	F	A	C	R	O	No.	B*	Thds.†	B_1	J	K
PS- 8	1	¼	⅛	5	2¾	2	¼	⅞	¾	⅛	⅛	8	⅜	24	1 1½₂	1⅜₄	1
PS- 9	1⅛	¼	⅛	5	2¾	2	¼	⅞	⅞	⅛	⅛	8	⅜	20	1 1½₂	1⅜₄	1
PS-10	1¼	⅝₄	⅛₂	6	3	2½	⅝₄	1 ⅜₄	1 ⅜₄	⅛₂	⅛	8	⅜	20	1⅜₄	1⅜₄	1 ⅜₄
PS-11	1⅜	⅛	⅜₂	6	3	2½	⅝₄	1 ⅜₂	⅛	⅜₂	⅛	8	⅜	20	1 1⅛₄	1⅜₄	1 ⅜₄
PS-12	1½	⅜	⅜₂	6	3½	2½	⅜₂	1 ⅜₄	⅞	⅜₂	⅛	8	½	20	1⅞	1⅜₄	1¼
PS-13	1⅝	⅜₄	⅜₂	7	3½	2¾	⅜₂	1 ⅜₄	1	⅜₂	¼	8	½	20	2	1½	1½
PS-14	1¾	⅜₄	⅜₂	7	4	2⅞	⅜₂	1 ⅜₄	1 ⅛₄	⅜₂	¼	8	⅝₄	18	2 ⅛₄	1½	1½
PS-15	1⅞	½	¼	8	4	3½	⅜₂	1 ⅜₄	1⅛	⅜₂	¼	8	⅝	18	2⅛	1 ⅝₄	1½
PS-16	2	½	¼	8	4⅜	3½	⅜₂	1 ⅜₄	1 ⅜₄	⅜₂	⅛	8	⅝	18	2½	1 ⅝₄	1¾
PS-18	2¼	⅝₄	⅛₄	9	5¼	3⅞	⅜₂	1½	1 ⅝₄	⅜₂	⅛	8	¾	16	2 1⅜₄	1⅞₄	1¾
PS-20	2½	⅝	⅜₄	10	5½	4¼	⅝	1½	1 ½₄	⅝₄	⅛	8	⅞	16	3⅛₄	1⅝₄	2⅛
PS-22	2¾	¾	⅜	11	6	4⅝	1 ⅛₄	1⅝	1 ⅝₄	⅜	⅛	8	1	14	3⅜	1⅝	2 ⅛₄
PS-24	3	¾	⅜	12	6¼	5	¾	1⅝	1 1⅜₄	⅜	⅜₄	8	1	14	3⅝	1 ⅜₄	2½
PS-26	3¼	⅞	⅜₄	12½	6⅝	5⅛	1 ⅜₄	1⅝	2½	⅜₄	⅜₄	8	1	14	3⅝	2 ⅛₄	2⅜₄
PS-28	3½	⅞	⅜₄	14	7	6	⅞	2	2⅝	⅜₄	⅜₄	8	1	14	4¼	2¼	3 1⅛₄
PS-32	4	1	½	18	8¼	6¾	1	2	2⅞	½	½	12	1	14	4⅝	2⅝	2⅝

*Figures in column indicate bolt diameters. Use clearance drill of same nominal size for holes *B*. All dimensions in inches
†All threads to be S. A. E. pitch, U. S. form. (See S. A. E. Handbook, Vol. I, sh. 4)
NOTE—P indicates propeller-shaft; S, steel; and figures, number of eighths of an inch in shaft diameters

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TABLE IV—DIMENSIONS OF FLANGE CAST IRON COUPLINGS (REVERSE GEAR AND PROPELLER-SHAFT)

Size No. (See Note)	A	KEYWAY		FLANGE				PILOT				HUB									BOLTS, S. A. E.			
		W	A	D	E	F	J	P*	P_1	Pc	Y	H	L	L_1	N	B_1	B_2	R	O	A	No.	B†	Thds.‡ per Inch	C
RPC- 6	¾	3/16	3/32	4	2 13/32	½	½	1 5/8	1/16	1/16	1/64	2	2	...	1/16	1 1/16	...	¼	⅜	⅜	4	⅜	24	3
RPC- 7	⅞	3/16	3/32	4	2 13/32	½	½	1 ¾	1/16	1/16	1/64	2	2	...	1/16	1 3/16	...	¼	⅜	⅜	4	⅜	24	3
RPC- 8	1	1/16	1/16	4	2 13/32	½	½	1 ⅞	1/16	1/32	1/64	2	2	...	1/16	1 3/32	...	¼	⅜	⅜	4	⅜	24	3
RPC- 9	1 1/16	1/8	1/16	5	3 1 13/32	⅝	5/16	2	1/16	3/16	1/32	2 1/16	2 1/16	...	1/8	1 1 3/32	...	5/16	3/32	1/16	6	3/16	20	4
RPC-10	1 1/8	3/16	3/32	5	3 1 13/32	⅝	5/16	2	1/16	3/16	1/32	2 1/16	2 1/16	...	1/8	1 3/16	...	5/16	3/32	1/16	6	3/16	20	4
RPC-11	1 3/16	3/16	3/16	5	3 1 13/32	⅝	5/16	2	1/16	3/16	1/32	2 1/16	2 1/16	...	1/8	1 1 5/16	...	5/16	3/32	1/16	6	3/16	20	4
RPC-12	1 1/16	3/16	3/16	6	4 1/16	⅜	⅜	2 1/16	1/16	3/16	1/32	3	...	4	3/16	1 1 5/16	1 1/16	3/16	3/16	1 1/16	6	3/16	18	5
RPC-13	1 1/16	5/16	3/32	6	4 1/16	⅜	⅜	2 1/16	1/16	3/16	1/32	3	...	4	3/16	2 1/16	1 1/16	3/16	3/16	1 1/16	6	3/16	18	5
RPC-14	1 3/16	5/16	3/32	6	4 1/16	⅜	⅜	2 1/16	1/16	3/16	1/32	3	...	4	3/16	2 1/16	1 1/16	3/16	3/16	1 1/16	6	3/16	18	5
RPC-15	1 1/16	1/8	1/16	7	4 1/16	1 5/16	5/16	3	3/16	1/16	1/32	4	...	5	1	2 1/16	2	5/16	5/16	1 1/16	6	1/16	18	5 1/16
RPC-16	2	1/16	1/16	7	4 1/16	1 5/16	5/16	3	3/16	1/16	1/32	4	...	5	1	2 1/16	2	5/16	5/16	1 1/16	6	1/16	18	5 1/16
RPC-18	2 1/16	1/8	1/16	8	5 1/16	1 1/16	1/16	3 1/16	1/16	3/16	3/64	4 1/16	...	6	1 1/16	2 1 5/16	2 1/16	1/16	1/16	1 1/16	6	1/16	16	6 1/16
RPC-20	2 1/16	1/16	1/16	8	5 1/16	1 1/16	1/16	3 1/16	1/16	3/16	3/64	4 1/16	...	6	1 1/16	3 1/16	2 1/16	1/16	1/16	1 1/16	6	1/16	16	6 1/16
RPC-22	2 3/16	1/16	1/16	9	6 1/16	1 3/16	1/16	4 1/16	1/16	3/16	3/64	5 1/16	...	7	1 1/16	3 1/16	3	1/16	1/16	1 1/16	6	1/16	14	7 1/16
RPC-24	3	3/16	3/16	9	6 1/16	1 1/16	3/16	4 3/16	1/16	3/16	3/64	5 1/16	...	7	1 1/16	3 1/16	3	1/16	1/16	1 1/16	6	1/16	14	7 1/16

*Pilot tolerance, +0.000, −0.001 in. per inch of diameter. Pilot recess tolerance, +0.001, −0.000 in. per inch of diameter
†Figures in column indicate bolt diameters. Use clearance drill of same nominal size for holes *B*
‡All threads to be S. A. E. pitch, U. S. form. (See S. A. E. Handbook, Vol. I, sh. 4)
(T. D.) = Tooling dimensions. All dimensions in inches
NOTE—R indicates reverse gear; P, propeller-shaft; C, cast iron; and figures (6, 7, 8, etc.), number of eighths of an inch in the shaft diameters

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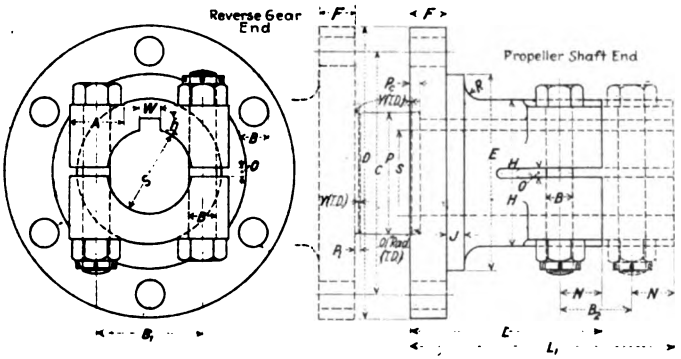
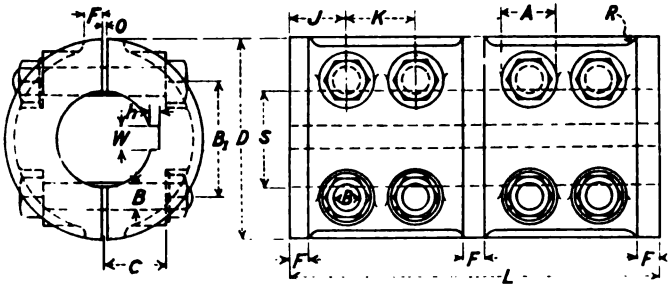


CAST-IRON REVERSE COUPLING

FLANGE CAST IRON COUPLING

(Reverse Gear and Propeller-Shaft)

The couplings shown in the accompanying drawings are for low power engines and may be made from a good grade of cast iron.



SHAFT COUPLING

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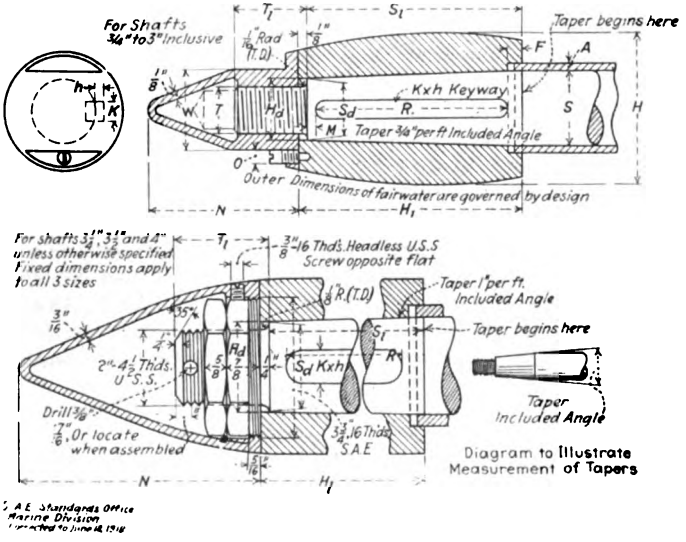
All bolts shall have S. A. E. standard threads as shown on Data Sheet 4, Vol. I, S. A. E. Handbook. (Safety sleeves or covers for these couplings are being considered by the Marine Division)

Propeller Mounting—For use in salt water, bronze or other non-corrosive metal is recommended for all parts of hub, shaft and fairwater assembly.

When bronze is used a tensile strength of 60,000 lb. per sq. in. and an elongation of 28 per cent in 2 in. is recommended.

The chemical composition of a bronze conforming to these physical properties is as follows: Copper, 62 per cent; tin, 1 per cent, and spelter, 37 per cent. A bronze having similar physical characteristics is commonly known as "Tobin bronze."

The small end of the shaft and hub tapers (S_d and H_d in the drawings and table) are the governing dimensions which insure proper assembly of varying hub lengths on the shaft and clearance between the end of the shaft taper and the bottom of the shaft nut.


For Shafts 3/4" to 3" Inclusive
Taper begins here
Kxh Keyway
Outer Dimensions of fairwater are governed by design
For shafts 3 1/2", 3 1/2" and 4" unless otherwise specified Fixed dimensions apply to all 3 sizes
3/8"-16 Thds. Headless U.S.S Screw opposite flat
Taper 1" per ft. Included Angle
Taper begins here
2"-4 1/2 Thds. U.S.S.
Drill 3/16"
3 1/2"-16 Thds. S.A.E.
Taper Included Angle
Diagram to Illustrate Measurement of Tapers

S. A. E. Standards Office
Marine Division
Corrected to June 12, 1918

PROPELLER MOUNT

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DISCUSSION AT STANDARDS COMMITTEE MEETING

W. S. HOWARD (M. S. A. E.): The Marine Division, after some nine months' work, has succeeded in adopting some standard flanges for coupling the reverse gear to the engine, and for coupling the propeller-shaft to the reverse gear; also for extension shafts having a clamp coupling, and for propeller mounts. The Government was consulted concerning the propeller mount and couplings and furnished blueprints for sizes from ¾ to 4 inches. The only requirement for the engine flange is that the manufacturer shall conform to the outer dimensions of the flange, which cover the pilot, the bolt circles, the number of bolts and their diameter.

The reverse-gear maker has to conform to the resulting outline, and as there are two forms of mounting, one flanged and one tapped in, provision is made for both types. An anchorage is provided on the flange for the head of the bolt; the pilot is also on the driving end.

It is the Division's recommendation that the reverse-gear tail-shaft always have a steel flange, mounted with a taper, as illustrated. Bolt circles and bolt sizes have been determined for the larger horsepowers, hence it is possible that in looking over these tables it might appear that some of the smaller sizes have larger bolts than are actually needed. The difficulty in some bolt shops without good machine facilities of turning tapers and mounting the propeller-shaft has been overcome. By substituting a cast-iron coupling for low horsepower two-stroke cycle engines the flanges will correspond to the steel flanges in a given series; instead of using set screws to keep the shaft in place when backing up, the bolts are so arranged that they will cut slightly into the shaft.

The recommended practice for clamp couplings for extension shafts is a manganese bronze, or bronze No. 29, S. A. E. specification. In this case likewise the bolts cut into the shaft to keep them from withdrawing. A substitute cast iron coupling is suggested for low horsepower.

Propeller Mount

The propeller-mount practice, as illustrated, conforms to the Navy practice with one or two changes, which have been agreed to on both sides.

Tolerances were considered first for the taper pro-

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peller-mount, also for the pilot, but as most of this standards work is new to the Division, we are not trying to do too much at first. I believe it will be possible, after a little, to give bolt tolerances and pilot tolerances. At present, chamfers are given; also the tool chamfer is marked "T. D." in the corners. Later the Division will standardize the parts of the reverse gears between flanges, but we have not reached that point yet.

THIRD REPORT OF TIRE AND RIM DIVISION

(*As adopted by the Society*)

As there was no quorum present at the last meeting of the Tire and Rim Division, which was held Dec. 10, 1917, in Detroit, the minutes were sent to the members of the Division for approval. The replies received indicated that the following subjects, which are now submitted for acceptance, were approved for standardization:

Solid-Tire Demountable-Rim Equipment—It is recommended that the demountable solid-tire equipment shown on War Department drawing T-101 be offered for standard service and for use only when demountable equipment is desired.

Types and Dimensions of Pneumatic-Tire Rims—It is recommended that for standard practice straight-side type of rims be used for all sizes, except for 3-in. and 30 by 3½-in., when the clincher type only should be used. (This notation is to be added to S. A. E. Data Sheet 8g).

Profile, Sizes and Types of Rims—It was recommended that S. A. E. Data Sheets 8f and 8g be combined, so as to make a more comprehensive sheet, with the subjects listed as follows:

(a) Drawings of rim contours.
(b) Table of sizes (same as shown on Data Sheet 8g).
(c) Routine tests (same as shown on Data Sheet 8f).
(d) For standard practice use straight-side type of rim for all sizes except 3-in. and 30 by 3½-in., when the clincher type only should be used.

Pneumatic Tires for Motor Trucks—It is recommended that the straight-side section be standardized with rim profiles according to the recommendation of the Tire and Rim Association for 6, 7, 8, 10 and 12-in. rims, and

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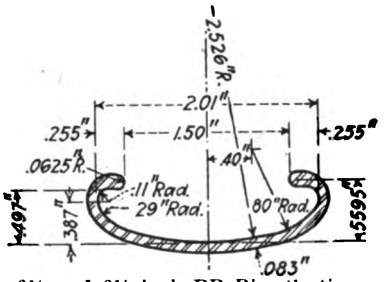
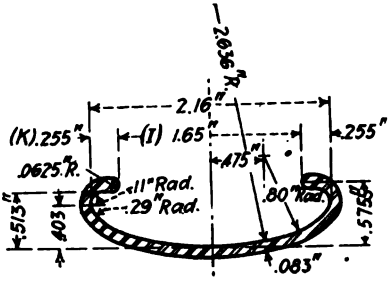
that the rim base-diameter be 24 in., as shown in the following table:

Nominal rim size, in.	Rim widths (width tire seat)	Tire sizes for 24 in. rim dia.
6	4.33	36 x 6
7	5.00	38 x 7
8	6.00	40 x 8
8	6.00	42 x 9
10	7.33	44 x 10
12	9.00	48 x 12

Motorcycle Rims—In view of the fact that the lighter types of motorcycles make use of the BB and the heavier type of the CC section rim, it is recommended that these two types be standardized as shown in the accompanying drawing.



2¼ and 2½-inch BB Rim Section



2¾ and 3-inch CC Rim Section

DISCUSSION AT SOCIETY MEETING

J. E. HALE (M. S. A. E.): The Automotive Wood Wheel Manufacturers' Association has had the question of wood felloe widths under discussion, manufacturers

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being invited to present their views. It was decided that the subject would need more attention before being finally settled. Because of this I think this subject should be referred back to the Division.

I offer an amendment that in the heading "Solid Tire Demountable Rim Equipment For Military Trucks" the words "for military trucks" be omitted and that in the sentence following the word "war" be left out. While this applies primarily to equipment in military service, it can apply equally well to equipment in commercial service. I think it should be a commercial as well as a military standard.

(The amendment was voted on and accepted)

SECOND REPORT OF LIGHTING DIVISION

(As adopted by the Society)

Headlamp Illumination—It is recommended that headlamps be so arranged that no portion of the direct reflected beam cone of light, when measured 75 ft. ahead of the headlamps, shall rise above 42 in. from the level surface of the road on which the vehicle stands, under any condition of loading; nor shall any portion of the direct-reflected beam cone of light rise, beyond the 75-ft. distance, more than 12 in. above the center of the headlamp.

To avoid a rise in the beam cone of light to an objectionable height beyond the 75-ft. distance, particularly in cases where headlamps are set relatively low on the car, the Division recommends that the word "at" be changed to "beyond" in the previously accepted recommendation. The direct and reflected rays (other than those direct-reflected rays from the parabolic reflector, which constitute the beam cone) are of such relatively small value that they have little influence.

Headlamp Bulbs—The Division has previously recommended that the focal length of electric incandescent lamps for automobiles be 1¼ in. In view of this and the fact that the G-16½ bulb is used ordinarily in electric vehicles, the 1¼-in. focal length is now recommended in the electric incandescent lamps for both gas and electric vehicle headlamps.

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BUSINESS SESSION OF SOCIETY MEETINGS

ADDRESS OF PRESIDENT

GEORGE W DUNHAM

With this, the Thirteenth Annual Meeting, the most eventful and perhaps the most important year in the history of the Society will be concluded. It has been a year not only of rapid growth but of remarkable accomplishments. It has been a year of attainment and patriotic service which can scarcely be duplicated if indeed it can be approached in the annals of any engineering society in this country, which rightfully boasts of the greatest engineering achievements of any nation. As a result the Society has attained recognition on the part of the Government not accorded to any other engineering organization, and has become in every sense of the word a national organization of which we as members may justly be proud.

Aside from the splendid work done by the Society in cooperation with the Government, perhaps its greatest attainment for the year has been the consummation of the plan for the unification of the engineering activities of all the automotive industries. This important accomplishment will, I am sure, have a lasting effect of the greatest good on the history of the Society. The word "automotive" has now become almost as general as the word "automobile," and the coordination of the automotive engineering activities of the industries concerned is being reflected in a closer unification of the industries themselves. I shall not attempt to forecast the future, but marked benefits seem certain to result from the closer cooperation of all branches of the automotive industry.

The growth of the membership of the Society during the past year has been quite in keeping with its rapid expansion in other directions. The membership on January 1, 1917, totaled 2120, while on January 1, 1918, the Society had 3119 members on its roll, a net increase for the year of 999 members or no less than 47 per cent, the greatest increase during any similar period in the history of the organization While this growth is due in part to the broadened scope of the work of the Society, in absorbing the engineering membership of four other organizations, great credit is due to the efforts of the

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Council, our Membership Committee and the Sections, which have given untiring efforts to the recruiting of new members in the older as well as the newer fields of endeavor. In addition, the Council, in cooperation with the Membership Committee, has laid plans for systematic membership increase work during the coming year, which is expected to bring in many strong men engaged in the automotive industries who have not heretofore fully appreciated the value of the work the Society is doing.

Committee Activities

In financial matters the Society has grown in proportion to its other activities. Details in this connection will be given in the report of the treasurer. Suffice it to say that although our expenses have increased considerably our income has fortunately increased almost correspondingly, and the Council has found it possible to add to what we hope will be a permanent investment by the purchase of twenty-four thousand dollars worth of first and second issue Liberty Bonds. For this accomplishment much credit is due to the wise counsel of the Finance Committee.

As a natural result of our broadened activities, more meetings have been held than during any similar period in the history of the Society. Besides the highly successful and very well attended annual and semi-annual meetings, three special meetings were held, these being the first aeronautic session in connection with the Aeronautic Show, and the first and second tractor sessions held in connection with the Kansas City Tractor Show and the Fremont Tractor Demonstration respectively. All were well attended and resulted in great good to the Society and the industries involved, as well as to those members who were able to attend. At the semi-annual meeting of the Society in Washington the attendance was so great that an overflow session had to be held, while at the banquet that followed the meeting the entire capacity of the largest dining hall in Washington was more than filled. The success of these meetings was due in large part to the splendid work done by the Meetings Committee, to which the Society owes a great debt of thanks.

In standardization matters the Society has done conspicuous service for the Government in the prosecution of its war program. The work done by the Truck Division of the Society was effective. The Government plan of establishing designs of war trucks involved considerable design work extending into the major units of

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the truck, a thing never contemplated in the standardization work of the Society. The Government put a large number of S. A. E. members to work, and the result has been the design of three sizes of entirely new trucks. The achievement is one that has not been accomplished by any of the other allied powers and one which is bound to have a decided and beneficial effect upon the prosecution of war.

Aeronautic Standardization

Although the Society has not been so intimately connected with the furthering of the Government aeronautic program, it has nevertheless done splendid work in the creation of aeronautic standards, and the Government has members of the Society to thank in large part for the design and development of the Liberty engine, an accomplishment that could hardly have been carried forward except for the missionary work the Society has been doing for years along standardization lines. In parallel with this development the Society has also through its Aeronautic Division completed the standardization of a number of parts entering into the construction of the modern airplane, and this work has been so well done that it is being incorporated substantially in the standards adopted by the International Aircraft Standards Board (on which the Society is officially represented) and the Aviation Section of the Signal Corps.

Work of a similar nature has been done by the newly created Motorcycle Division of the Society Standards Committee. Although the program of the Government in connection with motorcycle standardization is not yet as fully developed as that of truck standardization, a similar procedure is likely and seems certain to be quite as successful. In this also Society members are doing the bulk of the work.

For patriotic reasons the major work of the Society in standardization, as in other lines, has properly been that done in the interest of the Government. This work will, however, have a decided effect upon automotive industries after the war; and it is not by any means exclusively a war emergency matter, nor has it been done to the exclusion of other important standardization work. I cannot attempt to enumerate here all the standards created, but many of the divisions have done work of great credit to the Society. The Chairman of the Standards Committee and all the members thereof are deserving of our thanks and appreciation.

During the year no fewer than eleven new divisions of

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the Standards Committee have been created. This amplification of activity covers fully the newer fields of automotive engineering, and is bound to have a beneficial effect upon the industries concerned. In order to show the extent of our work, exhibits of parts made up to S. A. E. standards have been made at automobile, aeronautic and tractor shows, and similar exhibits are contemplated for several of the trade shows to be held within the next few months.

The prosecution of the standards work has, of course, involved considerable expense and could hardly have been carried on except for the generous financial support accorded by the National Automobile Chamber of Commerce, the Motor and Accessory Manufacturers' Association, the Aero Club of America, and the National Association of Engine and Boat Manufacturers. We believe these organizations feel well repaid by the results accomplished. I desire to take this opportunity to thank them publicly for their support.

Work of the Washington Office

The cooperation mentioned under the head of standards work is not by any means the only way in which the Society has been of assistance to the Government. Prior to the declaration by this country of war on Germany the Council decided upon the establishment of a Washington office. The function of this office has been to render the most effective assistance possible to the Government and the members along automotive engineering lines. One of the important things accomplished has been the taking of a census of the entire membership with a view to acquainting the Government with the qualifications of the members. The large number of blanks returned as a result of this census have been carefully classified according to the experience of the members, and have been made available to those branches of the government service requiring men in either civilian or commissioned capacities.

In large part as a result of this something over one hundred twenty-five members of the Society have received commissions and are doing engineering work of the greatest importance, and an almost equal number are serving in civilian capacity, while many more are giving a portion of their time to work in an advisory capacity. The work being done includes nearly every branch of design, construction, inspection, development, repair and operation of automotive apparatus. I shall not attempt to list the numerous activities involved, but

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one of the most important chapters in the history of the Society is certain to be that having to do with the splendid cooperation of our membership with the Government in the prosecution of the war.

The activities of the Sections of the Society have kept pace with those of the parent organization. The year has seen the establishment of two new sections, one in Buffalo and one in Minneapolis, both of which have held meetings of importance. The Sections Committee has done excellent work in correlating the activities of the sections, and the prospects for the future are bright. The work of the sections is properly regarded as one of increasing importance and it is to be hoped that members in all localities where sections are established will give them hearty support. In localities where there are few members, it is planned to hold occasional meetings which it is hoped will ultimately increase the interest in the work of the Society to such an extent as to warrant the establishment of sections.

New S A E Periodical

With the growth of the Society a larger and somewhat more ambitious monthly periodical became desirable, and we are now publishing what is known as THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS, a periodical which has received the highest praise not only from our members but from other well-qualified judges of technical publications. It should be realized, however, that the Society publications will necessarily be what the membership makes them. Each member should therefore give the Society the benefit of his experience by the preparation of papers on automotive engineering subjects; these should include original research data and design and engineering data of all descriptions that will benefit the profession and reflect credit upon the Society through which they are published.

In order to carry on the enlarged and ever-increasing work of the Society the Council has considered it desirable to secure larger quarters, and has therefore arranged with the management of the Engineering Societies' Building for the lease of its entire sixth floor, except one inside room not at present required. The Society will thus be housed in quarters befitting its size and importance in the engineering field.

In order to handle the increased volume of business done by the Society it has been necessary for the Council to employ a larger office staff. The work of the Society is thus carried forward through an organization created for

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the purpose without making undue demands upon the time of the members.

I have already referred specifically to work that the Society is doing in two of the newer phases of endeavor—the airplane and the farm tractor. Similar work is in progress in the marine field in which considerable standardization activity has been apparent for some time. Plans for a meeting of particular interest to those in this field have already been announced.

Similar activities have been planned for the members particularly interested in the stationary and farm engine field. A new division of the Standards Committee has been created to undertake standardization in this branch, and other work corresponding to that already accomplished in the older lines of endeavor is contemplated.

I should not feel my duty complete if I failed to mention a severe loss sustained by the Society during the year. I refer to the untimely death of the late Major Henry Souther, past-president of the Society and for some years chairman of its Standards Committee. His untiring efforts in the interest of the Society will long be remembered.

In closing let me say that the extraordinary results obtained during the year could not have been secured without the hearty support and untiring efforts of all the members of the Council, and the closest cooperation and efforts of our committees and the members of the Society in general. The good work must and I am sure will be carried forward through the officers of the incoming administration and the others chosen to represent the members on next year's Council, and I am sure that even greater success will crown their efforts for the Society, the automotive industries and democracy.

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TREASURER'S REPORT

The Treasurer's report for the eight months ended May 31, 1918, is as follows:

BALANCE ON HAND, Oct. 1, 1917

Savings Account	$7,000.00	
Checking Account	4,985.95	
Cash in Office	390.56	
		$12,376.51

Receipts

Initiation Fees and Dues	$55,651.61	
Pin, Binder and Folder Sales	3,220.85	
Publication and Other Receipts	73,801.48	
	$132,673.94	
Less Cash Discount	605.63	
Total Receipts		$132,068.31
		$144,444.82

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Total Amounts Brought Forward		$144,444.82
Disbursements		
Salaries	$27,962.28	
Pins	3,070.15	
Postage	4,364.57	
Publications	24,731.65	
Rent	5,059.12	
Meetings	12,282.39	
Telephone and Telegraph	853.93	
Stationery and Supplies	2,656.32	
Express	752.22	
Furniture and Fixtures	1,686.19	
Standards Committee	6,974.76	
Convention Reporting	296.83	
Investment	22,000.00	
Unclassified	10,574.53	
	$123,264.94	
Less Discount	749.67	
Total Disbursements		$122,515.27
		$21,929.55

BALANCE ON HAND, May 31, 1918

Checking Account	$10,729.55	
Petty Cash	450.00	
Certificate of Deposit	6,000.00	
Office Manager's Account	4,750.00	
		$21,929.55

May 31, 1918

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REPORT ON CONSTITUTIONAL AMENDMENTS*

The Constitutional Committee of the Council has been requested to give consideration to suggested amendments to the Constitution covering

(1) Provisions for grading the membership

(2) Provision for a foreign grade of membership

(3) Provision for the United Service grade of membership, and

(4) Provision for an additional administrative committee to be known as the Sections Committee.

It has been felt that a revision of the classification of active members is desirable to permit the broadening of the activities of the Society by means of increased membership. Several suggestions to this end have been presented for the consideration of the Committee.

The first is an amendment to Paragraph C 8 of the Constitution, which would more clearly outline the qualifications governing admission to Member grade; the second is to change the designation of grading from Member and Associate Member to Technical Member and Industrial Member, the idea being that there is a certain class distinction between Member and Associate that would be more clearly indicated by the suggested change and that would appeal to men of high standing in the industry who feel that their ability and contributions to the industry have not been fully recognized when they are assigned to Associate membership, but who are not possessed of the necessary qualifications in a technical way to admit them as Members.

The Committee feels, however, that this proposal would not cover the situation unless an amendment were made that would give the Industrial and Technical Members equal powers with respect to holding office and voting; this would result in there being no practical difference between the two grades of membership and therefore no reason for the proposed difference in name.

A third alternative is that all membership distinction

*The amendments incorporated in this report were submitted in writing at the annual meeting held January, 1918, in New York, being proposed by J. G. Utz and seconded by C. B. Whittelsey. In accordance with C 56 of the Constitution of the Society they were received, for presentation at the next semi-annual meeting.

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be eliminated among the active members, the division to be somewhat as follows: Members, Junior Members, Affiliate Members, etc.

This suggestion has certain advantages inasmuch as it would remove the necessity of the Council deciding or using any discretion other than the acceptability of an applicant for membership. It is open to some objection on the score of being a revolutionary departure from the precedents established in the conduct of practically all engineering organizations. It has therefore appeared to the Committee that the first alternative is the only one that it can at the present time present to the Society for consideration.

It is proposed that C 5 of the Constitution shall be amended to read:

The membership of the Society shall consist of Honorary Members, Members, Associates, Juniors, Departmental Members, United Service Members, Foreign Members and Affiliate Members. Honorary Members and Members are entitled to vote and to hold office. Associates, Juniors, Departmental Members, United Service Members, Foreign Members and Affiliate Members shall not be entitled to vote or to be officers of the Society, but shall be entitled to the other privileges.

It is proposed that C 8 of the Constitution be amended to read:

Member grade shall be composed of persons 26 years of age or over, who, by previous technical training or experience, or by present occupation, are qualified to act as designers or constructors of complete automotive vehicles or their important component parts; or to exercise responsible technical supervision of the production of materials germane to the construction of automotive vehicles; or to take responsible charge of automotive engineering work; or to impart technical instruction in automotive vehicle design and construction; or who, by reason of distinguished service or noteworthy accomplishment would, in the discretion of the Council, appear to be desirable additions to this grade.

The Committee feels that this amendment clearly outlines the qualifications for membership in the Society based on technical ability and still allows for the admission to full membership of men who by reason of their position in the industry are entitled to have a voice in the direction of its affairs.

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After C 10 the following shall be added:

United Service Membership shall be composed of engineers 26 years of age or over, engaged exclusively by the Federal Government, the qualifications for this grade being the same as those for Member grade.

Foreign Membership shall be composed of engineers 26 years of age or over, resident in countries other than the United States, Canada, Mexico or Cuba. The qualifications for this grade shall be the same as those for Member grade.

A member who has become eligible for either the United Service Grade or the Foreign Member Grade may, upon written request, be transferred to such grade and thereafter, beginning with the next dues period, pay the annual dues for said grade.

C 13 shall be amended to read:

The rights and privileges of every Honorary Member, Member, Associate, Junior, United Service Member and Foreign Member shall be personal to himself and shall not be transferable by his own act or by operation of law. Departmental Membership and Affiliate Membership also shall not be transferable.

C 15 shall be amended to read:

All original applications for election to the grade of Member, Associate, Junior, United Service Member or Foreign Member shall be presented to the Council, which shall consider and act upon each application, electing each applicant to the grade of membership to which in its judgment his qualifications entitle him. Two negative votes shall defeat an election.

C 20 shall be amended to read:

The initiation fee for membership in each grade and for Student Enrollment shall be as follows:

For Member,	$25.00
For Associate,	25.00
For Junior,	10.00
For United Service Member,	10.00
For Foreign Member,	10.00
For Student Enrollment,	None
For Departmental Member,	100.00
For Affiliate Member,	50.00

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C 21 shall be amended to read:

The annual dues for membership in each grade and for Student Enrollment shall be as follows:

For Member,	$15.00
For Associate,	15.00
For Junior,	5.00
For United Service Member,	10.00
For Foreign Member,	10.00
For Student Enrollment,	3.00
For Departmental Member,	None
For One Affiliate Member Representative,	15.00
For each additional Affiliate Member Representative,	10.00

C 22 shall have the following added thereto:

A United Service Member shall, upon discharge from the Service, promptly notify the Secretary of such discharge, and shall then be transferred to Member grade and shall pay the difference in initiation fee (provided he has not previously paid such fee) and thereafter beginning with the next fiscal period, shall pay the annual dues for Member grade.

A Foreign Member shall notify the Secretary upon change of residence to territory rendering him ineligible for Foreign Grade, or temporary change of abode to such territory for more than six months in a given dues period, and shall thereupon be transferred to Member grade and shall pay the difference in initiation fee for Member grade and thereafter, beginning with the next fiscal period, shall pay the annual dues for Member grade.

C 25 shall be amended to read:

Any Member, Associate, Junior, United Service Member, Foreign Member, or Affiliate Member, who shall leave the annual dues unpaid for three months shall not receive any publications of the Society until such dues are paid. Any Member, Associate, Junior, United Service Member, Foreign Member, or Affiliate Member who shall leave the dues unpaid for one year shall, in the discretion of the Council, cease to have any further rights in the Society and be stricken from the rolls of membership. The resignation of a member, whose account with the Society is not fully settled, can be accepted only by vote of the Council.

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C 26 shall be amended to read:

The Council may temporarily suspend the annual payment of dues by any Member, Associate, Junior, United Service Member or Foreign Member, whose circumstances have become such as to make it impossible for him to pay the dues, and may under similar circumstances waive the whole or part of dues in arrears. Such action shall be taken only upon the written application of the Member, Associate, Junior, United Service Member or Foreign Member, signed by three other members in good standing, or evidence satisfactory to the Council that the action is for the best interests of the Society.

It is the intent of the amendments that the United Service and Foreign Member grades be strictly limited to persons possessing qualifications for full membership, because the dues and initiation fees provided mean that these grades will be carried at a loss to the Society. It is, however, felt that there is a real field for the Society to do excellent work and to put its information into the hands of desirable persons by this method.

C 45 shall be amended to read:

The President shall, within thirty days after taking office, appoint, from the individual membership of the Society, the following Committees, designating the Chairmen thereof:

Finance Committee, consisting of five members,
Meetings Committee, consisting of five members,
Publication Committee, consisting of five members,
Membership Committee, consisting of five members,
House Committee, consisting of five members,
Sections Committee, consisting of five members,
Tellers as required by the By-Laws.

There shall be also a standing committee of the Society called the Constitution Committee, consisting of three members, one of whom shall be appointed by the President within thirty days after taking office, for a term of three years. The member who shall have but one year yet to serve shall be the chairman.

At present, Section matters are handled by a committee consisting of a member of the Council, the chairmen of the various Sections and a member of the Society, who is a representative of the New York Office, acting as secretary. It is felt that the importance of the work of the Sections makes it desirable that these matters be handled by a standing committee of the Society appointed by the President under authority of the Constitution.

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TYPES OF MILITARY AIRPLANES

By Col V E Clark*

The military types of airplanes, as influenced by military functions and as influencing the design of the engine that goes into the plane, will be the subject of this brief discussion.

The Allies have run to more types than the enemy. One of the main reasons for this is that the Allies have listened more to the demands of their fliers, who have exerted much influence on plane and engine design. Because of this, standardized production has been sacrificed to individual requirements.

It will be America's part in the aircraft program to select those types that best lend themselves to big production—to look ahead six or twelve months in an endeavor to anticipate the advance requirements, and then design those types in such a way that they can be built on a large scale. Standardization means fewer types and greater numbers of machines. We must select perhaps five or six types, develop them, and *produce* them by the thousands.

We should select for standardization from the following types:

(1) Airplanes of Observation.
(2) Airplanes of Combat and Pursuit.
(3) Airplanes of Destruction and Harassment.
(4) Special Types.

None of these types is new. All can be developed to a far greater degree of efficiency than those now being used. Each type has a different function. Airplanes of observation must aid the army on the ground in successfully performing its operations. Those of combat and pursuit must prevent enemy aircraft from doing damage in any way. Those of destruction and harassment will inflict direct damage on the enemy. With special types we have little concern. Of chief concern to the Aircraft Board—to America—in fact, to the world, are those planes that will inflict direct damage on the enemy. They are the bombing machines, upon which depends a substantial part of the measure of victory; of that I am convinced.

The need for these being vital and well recognized, and the hour of their initial production being at hand, I shall

*Airplane Engineering Division, Signal Corps, U. S. Army

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confine my remarks to a limited discussion of their possibilities.

Bombing operations provide practically all the real damage that it is at present possible for an airplane to inflict upon the enemy. In comparison with these the slight damage caused by bringing down his airplanes, each containing one or two men only, or shooting up his trenches or truck trains, is negligible.

The enemy should be harassed continually from the air. Two classes of bombers should be employed—day bombers and night bombers.

Use of Day Bombers

Day bombers will be used almost entirely for the demoralizing effect on the enemy of perpetual anxiety, except in a few special cases where vulnerable objectives are difficult to find by night.

The primary military functions of the day bomber are:

(1) To destroy important points, such as small objects difficult to find, headquarters, small ammunition "dumps," small storehouses containing munitions or supplies, small railway junctions, and small aerodromes.

(2) To bomb such communities as is considered desirable, especially factories and factory towns.

(3) To conduct long-range reconnaissance, strategical reconnaissance, reconnaissance by staff officers or with camera.

(4) To do special photographic work so far beyond the lines as to necessitate great altitude, demanding a camera of great focal length and therefore unusual size and weight.

The primary requirements for such an airplane in order that it may effectively perform its military functions are:

(a) Ability to protect itself effectively against all hostile aircraft, which demands good speed at altitude, strong climbing ability, powerful and reliable armament, and a satisfactory degree of "handiness."

(b) A reliable powerplant.

(c) A powerplant with good fuel efficiency.

(d) Large bombing capacity. I believe that, at the present time, it is not an economic proposition to send a trained pilot and a trained "bombardier" a great distance beyond the enemy's line unless at least 600 pounds of bombs are carried.

(e) Effective provision for accurate sighting and dropping of bombs.

(f) Ceiling sufficiently high to give the machine a good chance of escaping detection as it crosses the line.

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(g) Muffler for the exhaust arranged to be cut in and off at the will of the pilot.

(h) Two or three machine guns, one firing through the propeller disk and one or two with all-around fire, with good field to the rear.

(i) Provision to carry two men.

(k) A reliable compass.

Typical airplanes of the day bomber type are the DeHaviland-9 (British), with 300-hp. Fiat engine; the S. I. A. 7-B (Italian), with 200-hp. Fiat engine; the S. I. A. 9-B (Italian), with 600-hp. Fiat engine, and the Breguet 14-B2 (French), with 300-hp. Renault engine. The German Gotha twin-engine machine (two 260-hp. Mercedes engines), while rather too slow and too unhandy for the purpose, has done some service in bombing by day over London.

Night Bombers

The type designed for bombing by night must, in my opinion, be depended upon to inflict real material damage upon the enemy. I believe that the consistent employment of these machines in large numbers on every good moonlight night to bomb Germany's munition factories, factory towns, important railway junctions, large munition depots, the bridges across the Rhine, the Kiel Canal, important docks, submarine bases and certain cities, would end the war in a shorter period of time than is possible by any other means.

The primary requirements for these machines in order that they may effectively perform their functions are:

(a) Great bomb capacity.

(b) Reliable powerplant.

(c) Powerplant with good fuel efficiency.

(d) Proper degree of stability and ease of control to enable a pilot of ordinary ability to fly and land at night.

(e) Effective provision for accurate sighting, and dropping of bombs.

(f) Accurate compass and other instruments needed for navigation by night, with provision for reading conveniently.

(g) Provision for carrying two to five men. Probably the best practice is a crew of three, a chief pilot, a "bombardier," and one man at the gun forward or to the rear, as may be necessary, and to act as relief pilot.

The load of bombs that can be carried will depend upon the total power available at an altitude of 10,000 feet, and upon the distance of the objective (which will

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regulate the initial fuel supply). The ratio of total weight of airplane, with full initial load, to the total power available should be small enough to permit a ceiling of at least 11,500 feet, starting with full load. The powerplant will be divided into two or possibly three units. Suppose that two U. S. A. 12-cylinder engines be installed; if no device is incorporated to maintain the power constant with change in altitude, the total power available at 10,000 feet altitude should be about 450 horsepower. Suppose that the objective lies 155 miles beyond the lines, a bomb load of between 2000 and 2700 pounds can be carried, and the necessary initial ceiling obtained, provided the general design of the airplane be good.

Limiting Weight

The total weight of the airplane in pounds with full initial load should not be more than 22 times the number of horsepower available at 10,000 feet. The total weight should not be more than 5.63 lb. per square foot. The machine should have a possible horizontal speed, at an altitude of 10,000 feet of not less than 85 miles per hour. Starting with full load, the airplane should be capable of climbing to an altitude of 10,000 feet in not more than 27 minutes. For every 16 miles' increase in radius necessary to reach the objective, 100 pounds of bombs are sacrificed.

Typical airplanes of the night bomber type are the Caproni triplane (Italian) with three 273-hp. Isotta-Fraschini engines; the Handley-Page (British) with 320-hp. Sunbeam engines; and the Caproni biplane with three 210-hp. S. P. A. engines. The German Gotha, with two 260-hp. Mercedes engines, is typical.

The number of night-bombing airplanes built and supplied should depend solely upon the number of pilots available for this work. A far lower degree of flying skill is required to pilot a large, slow, night bomber than a fast, fighting machine, although more mature judgment is necessary. As a matter of fact, the number built and supplied will, in all probability, eventually depend upon steamship space for transportation, upon the hangar space at the aerodromes in France, and, possibly, upon the appropriation available.

Air Raids With Bombs

Consider, for example, a fleet of several hundred night-bombing airplanes, each carrying a ton and a half of bombs, flying from large aerodromes placed say 25 miles

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to the rear of the line. The fleet penetrates to Essen, for instance. Each machine finds its objective and drops ten 160-lb. bombs of the high-explosive type on the factories and forty 25-lb. bombs filled with poisonous gas and twenty-four 25-lb. bombs of the incendiary type throughout the factory town, and returns home.

In the existing phase of the war were our night-bombing airplanes of sufficient numerical strength, it would be no longer a matter of individual and isolated raids on selected places at which the maximum of injury could be inflicted, although even consistent bombing of factory towns would end the war surely and quickly, but rather a continuous and unrelenting attack on every point of strategical importance.

Depots of every kind in the rear of the enemy's lines would cease to exist; rolling stock and mechanical transport would be destroyed; no bridge would be allowed to stand for twenty-four hours; railway junctions would be subject to continuous bombardment, and the lines of railway and the roads themselves broken up nightly by giant bombs to such an extent as to baffle all attempts to maintain or restore communication.

In this manner a virtually impossible zone would be created in the rear of the enemy defenses, a zone varying from 100 to 200 miles in width. As soon as this condition had been brought about, the position of the defending force would be precarious, and would eventually become impossible. The defense would be strangled by uncertainty and lack of supplies of all kinds. Ultimately retreat would become impossible. The defending force would find itself in a state of siege under the worst possible conditions. Its position would be in the form of an extended line along which the forces of all arms would be definitely immobile, since the lateral communication would suffer no less than the lines from the rear. In short, a reign of terror would exist. Such a condition presents all the elements conducive to complete and irreparable disaster.

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REASONS BEHIND THE LIBERTY AVIATION ENGINE

By MAJOR JESSE G VINCENT*

Last spring the Government had before it the problem of deciding on an aircraft program. We heard at that time and we still hear that we should take the best engines available abroad and manufacture them in this country. The French and English engineers agreed that there was no engine then being built in this country that would be of any value at the front. They told us that an engine weighing about two pounds per horsepower was required, and that although such engines were not then a reality they were in process of development, and it was hoped that they would be built within a year.

In other words, last spring it was impossible to select an engine from abroad because the manufacturers there did not then have a type with which they were themselves satisfied. They were getting along the best they could with what they had, so it was finally decided to take all the best features embodied in the engines of both the Allies and the Germans and combine them into a composite model having no experimental features. It was desired to combine all the benefits of the experimental work done by the Allies and at the same time have the fewest possible number of types, a manufacturing method followed by the Germans.

The original scheme embodied the manufacture of a number of engines with interchangeable parts. The cylinders were to be varied from four to twelve, possibly using more later if it seemed desirable. At first it was decided to build an eight-cylinder engine of about 150 hp., but before that was completed we were asked to provide a twelve-cylinder engine giving over 300 hp., because more power was necessary.

We must remember that there was no engine in this country developing enough power to satisfy the requirements at the front. It was necessary therefore to provide the entire manufacturing facilities for one or more high-power aircraft engines. If ten or fifteen factories in this country had been asked to design engines of from 350 to 400 hp., at least ten or fifteen different kinds of engines would have been the result. Moreover,

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it would have been impossible for all the designers to have before them the latest information that we had available. It was decided therefore that it was absolutely necessary to design a completely new engine, one for which the drawings and specifications would be prepared by the Government, so that they could be handed to the manufacturers to produce the engines in quantity.

I want to acknowledge the hearty cooperation of the automobile engineers and manufacturers in doing the work. It was only on June 4 that Major Hall and I were ordered to produce ten sample engines. We had laid out at Washington just about what the design was to be. We then went back to Detroit and engaged a large force of draftsmen to prepare the rough drawings. These were sent out through the country and the first engine was produced in twenty-one days. The patterns and die for the crankshaft and everything else were produced in that time.

In designing the engine we took it unit by unit and considered first what would be the best design and second what would be the best manufacturing construction. We took the cylinders, the valve-operating mechanism, the crankcase construction, the crankshaft, the ignition, and handled them in this way.

The building of the twelve-cylinder engine of course was simple after the eight-cylinder had been finished, because we used practically the same parts except for the crankshaft and crankcase.

When the twelve-cylinder engine was finished, it was given a 50-hr. test; this was accomplished without making any changes and without making a single adjustment during the run. The engine was not operated at the maximum horsepower during the run because it was not designed to be run at such power on the ground. Since that time, however, we have found that it will run wide open an indefinite length of time on the ground. We have also found that by making a few minor changes it will stand over 400 hp. continuously in field service. It is designed for fighting machines intended to fly at altitudes of thousands of feet, and weighs about two pounds per horsepower.

When the sample engines were being built the Aircraft Production Board and the Signal Corps were arranging for facilities so that the engine could be manufactured in quantities. The equipment for manufacturing the steel cylinders is an example of the production methods. These cylinders are manufactured from tubes, and by a clever arrangement the end that is closed in and the small fissure

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that results from the closure of the end come just where the port is bored out, so that there is no bad metal there.

The production of the Liberty engine will not only be of great benefit during the war, but it will also do the whole automotive industry a great deal of good after the war. The engineers of the Society have been cooperating for years and, I believe, work together better than the members of any other engineering society in the world. Our factories have also cooperated to a large extent, but the work now going on in the automobile factories producing these engines is bringing them together to a marked degree. This applies not only to the plants making the engines, but also to the companies making the parts.

If a number of different aircraft engines had been designed in different factories, it would not have been possible to secure this cooperation. Each man would naturally have tried to make his own engine the best. The most important thing is the fact that with the standardized engine we get the benefit of everybody's cooperation on one article, and we can make that engine a great deal better than if any one company continued to make it for a year.

At present we have nothing but twelve-cylinder engines in production, and do not believe that anything else will be in production during the war, because we want all the power that can be used efficiently.

THE DISCUSSION

(The discussion following the remarks made by Colonel Clark and Major Vincent consisted merely of questions regarding the operating possibilities and construction of the Liberty engine. The answers by Major Vincent are given below, substantially in complete form.)

The cruising radius of the machines in which the Liberty engine will be used is entirely a matter of plane design and the load to be carried. The twelve-cylinder engine when placed in a small, fast machine will give a considerable cruising radius, provided the load carried is not great. On the other hand, two or more of the Liberty engines placed in a large machine with a heavy load will either decrease the power of the engines or lower the speed. The radius could easily be as great as 600 miles, that is 300 miles out and 300 miles back.

A number of problems have been discussed in connection with the ignition. The feeling on the other side of the water is that with the increased number of cylinders and the increased number of engine revolutions, the

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magneto is unquestionably the best. But we must have two sparks to a cylinder and we must have those sparks in such a way that they can be obtained separately. The whole design is a compromise, and weight is one thing on which we do not want to compromise too much. I believe that the ignition system on the Liberty engine is at least as good as anything that can be designed for it at present.

We will manufacture some engines with battery ignition. In the meantime we shall try out any system that looks at all good. I will say that no other system that has been offered to us has been as light, combined with equal reliability, as the one now in use.

The four-valve construction was tried on a Mercedes engine and immediately abandoned. Four valves are much harder to cool than two and should not be used until the limit is reached with two valves. Until a lift of one-quarter inch of the clear diameter is used, the valve is not really being operated efficiently. It would be foolish to use four valves in an engine that was not more than 4½-in. bore by 7½-in. stroke.

There is no reason why the Liberty engine cannot be operated satisfactorily at low speed, the slowest useful speed for aviation being 1200 r.p.m. It has no unusual valve-timing. The exhaust valve opens 52 deg. early and the inlet closes 45 deg. late.

The present ignition system weighs 29 lb., this including the battery. The test made of the Liberty engine on Pike's Peak showed that power was gained up to a compression ratio of 6 to 1, but that the same engine when brought down to the surface of the earth operated poorly.

Fuel is fed by pressure as in the ordinary method. Auxiliary tanks are provided, these to be used in case of emergency. The lubricating system is of the pressure type. A double-deck pump is used with three gears in the top deck, thus making two suction pumps. The oil is drawn out of the rear crankcase and from the front into a separate case. We have followed the system used by the Rolls-Royce Co., and well known on the other side, of employing a dry sump case, pumping all the oil into a little tank and then pumping it back underneath. Originally the connecting-rods were lubricated by scuppers on the crankshaft which caught the oil that was thrown off. That method was extremely economical, but with it a cold engine could not be started and full power secured.

Between 1200 and 1300 r.p.m. there is a noticeable vi-

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bration in the operation of the engine, depending to a certain extent on the propeller design. The block angle of 45 deg. has seemed to offset the ordinary calculated period of vibration.

The new Liberty engine is said to be 50 hp. more powerful than the Rolls-Royce and at the same time at least 150 lb. lighter in weight. The valves are set at angles of 30 deg. The radiator capacity is from ten to twelve gallons.

The Liberty engine originally weighed 786 lb., but some 15 lb. have been added by slightly increasing the weight of three or four parts, thus bringing the total up to about 800 lb. It has deevloped 400 hp. at 1625 r.p.m., and of course more power can be secured by running at higher speed. The speed is limited, however, because we cannot get an efficient propeller speed much above 1600 r.p.m.

The best French engines give between 200 and 250 hp. and weigh about 520 lb., these being of the geared type.

The geared engine is best for some types of work and the direct-drive for other types, but the latter engine is the more reliable, which is wanted, of course, above everything else. The manufacture will be concentrated at first on direct-driven engines, although investigations will be continued as to the possibilities of the geared type.

Two compression ratios are used at present for the Liberty engine. In the navy engines the compression ratio is 5 to 1 and in the army engines 5.4 to 1, or something under 18 per cent. The low compression is best for navy work, which consists largely of flying at altitudes of about 5000 ft. or less. It is desired to fly with a wide-open throttle most of the time, so that the navy engine should run well near the earth's surface. On the other hand the army engine is used for altitudes of around 10,000 ft. or more. The larger ratio is necessary therefore so that too much power shall not be lost in the air. It is believed that with the steel cylinder construction, which makes for satisfactory cooling, a still higher compression can be used. It is possible to run the engine up to as high as 6 to 1 compression ratio.

The steel cylinders are arranged so that the cooling water can be brought up close to the spark-plugs. The compression pressures have not yet been carried extremely high in the Liberty engine so that spark-plug trouble, as ordinarily known in aviation engines, has not been experienced.

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Use of Engines

The Liberty engine in production will be used mostly for fighting work, inasmuch as other satisfactory engines are being built in this country for the training work, and these will be continued in production. The only training planes in which Liberty engines are to be used will be a few fighting machines in the schools for the final training of pilots.

The airplanes in use at the front are seldom worn out. The only worn-out ones are possibly those used in the training camps. When those at the front go out of service it is usually due to an accident of some sort. The average life of a combat machine is about two months, and that of the observation machines used in the Army something like three months. The light bombers probably have a much longer life.

Good facilities are now available in Dayton to carry on experimental testing, and it is planned to conduct a great deal of research work in the air, something that it has been almost impossible to do in this country heretofore.

A good deal of time has been wasted in designing self-starters. A self-starter is not desired on the plane for combat work because the weight can be used to better advantage in the form of bombs or some other useful things. It is impossible to stop the engine after leaving the ground; hence a self-starter is unnecessary. The Navy machines will have self-starters. Whether these are to be air-starters or electric-starters will be worked out later. There is no advantage in having self-starters in the ordinary single-seater or double-seater machines because there are always plenty of men on the ground to start them.

H. M. CRANE* (M. S. A. E.):—It was necessary to build an American engine not because we wanted the eagle to scream, or for patriotic reasons, but for considerations based on practical common sense. If we had imported engines and copied them here, in a year they would have been either American engines or Americanized engines. The only way we could get the industry of the country at work was to use a design that would meet factory ideas in the United States. It is difficult for one factory engineer to design an engine to be built in another factory in this country, without knowing that factory. It is almost impossible for any foreign engineer to design for an American factory,

*Chief Engineer, Wright-Martin Aircraft Corporation

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If we had used an American factory to build a foreign engine, the time for getting it well into production would have been as long as if the engine had been of American design. In addition to this, the production would never have been half what we can expect for an engine designed in this country, with full knowledge of American factory practice and personnel back of the design.

Early last summer, when I read the newspaper reports that an engine had been designed in three weeks that would lead the world in aviation, I was much startled. When I had the privilege of coming closer to what had been done I found that the newspaper story was only the final printing of the message. What had really occurred was a year or two years of painstaking development of what had been proved good on the other side, thinking always in terms of American methods of manufacturing. The result was that in a short time—so short that it seemed almost impossible—the knowledge thus acquired was transferred to drawings and from drawings transferred to material and to an engine in operation.

For nearly two years, I had some experience with a foreign engine, designed I consider by the best production engineer abroad. For that reason it has been immensely successful on the other side, yet for this country its manufacture resulted in difficulties that had to be met by adding to our own plant facilities that we should have been able to depend upon in the industry at large.

For a small output we could add these special facilities, but when the Government found it necessary to mobilize the industry as a whole, such a course of action would have been impossible. Instead of our being of real help sooner than most of our Allies expected, the war might be over before the different factories would learn how to do things in the new way. Another difficulty in manufacturing an engine that it is necessary to copy is that factories do not know the history of the unsuccessful experiments which have resulted in the finished product available, and the problem is made much harder. An engine may have been developed in a factory in which certain machine tools were available and may even have been built depending upon certain foremen who knew how to do their work extremely well. The Mercedes engine is an illustration of this. A number of attempts have been made to copy it, but I do not know of any copy that can be compared with the original in efficiency or usefulness. The manufacturers who have finally secured a fine product as a result of copying the Mercedes

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engine have, in doing so, altered their processes and the design materially. That is undoubtedly what would have happened to us if we had started on a foreign design, for the foreign designs most successful today are not inventions; they are not something startlingly different. We must thank the Germans for showing us that an aviation engine can be of the simplest form, and that the simple construction and form are the best, for thus reliability and easy production are secured.

Our knowledge of German experience was a help in going ahead in this country. The Liberty engine is the embodiment of the idea of a small number of parts, simplicity of parts and compactness, all of which make for light weight and reliability.

As Major Vincent has said, the cooperation that has arisen in developing the Liberty engine is a matter of tremendous importance in the fighting of the war and will continue to be afterward. It will make many new friends for all of us, and furthermore we shall know these friends in a way that we have never known them before. I believe that when we look back after two years all of us will feel that this Liberty engine is a rallying point, a central point in the industry, and that it will go a long way toward increasing the efficiency of our country in fighting and winning the war.

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CHASSIS DESIGN OF CLASS B MOTOR TRUCKS

By CORNELIUS T MYERS*

If there is an accomplishment any more representative of the spirit and constant endeavor of the S. A. E. than the design and building of the first Class B military motor truck it does not appear on our horizon at this moment. It represents one culmination of the ideals for which the Society has labored unceasingly, and that it is to be of service to our country at this time is a reward that few of us expected.

In connection with chassis designing I will give the sequence of events leading up to the starting of the work, and will then touch upon a few fundamentals, following with notes as to details.

On June 27, 1917 (the day after the close of the Semi-Annual Meeting of the S. A. E. in Washington), a large number of engineers representing truck builders and component parts makers met in Washington to discuss with representatives of the Quartermaster General's office, a military motor truck composed of standardized parts. These men divided up into groups, representing engines, axles, transmissions, chassis, etc., and after some discussion it was decided that the job could be done. A tentative layout for the Class A chassis was discussed. A layout for this Class A truck using the parts suggested by the parts makers group was prepared and submitted at the next meeting, which was held July 9 and 10 in Columbus. The layout was discussed and a few changes recommended, which were incorporated and submitted July 20 and 21 for the second Columbus meeting.

A layout of the Class B chassis was made and quite fully discussed. The Quartermaster General's office then took over the work, having decided that a truck standardized in all its parts should be designed. A number of the engineers who had been working on the chassis and parts designs were asked to go to Washington and complete the design under the supervision of the Quartermaster General's office. On July 30 these engineers reported in Washington ready to continue the work.

On August 27 the Schedule Committee on Design was organized and the work divided among parts groups, setting September 10 as the day on which the design should

*Consulting Engineer, Detroit

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be completed. Between July 30 and August 27 much preliminary work was done by the groups of parts builders but no progress could be made with the chassis design, as there were differences of opinion among the members of almost every group of parts designers.

Sample Trucks Started

The parts designs were finished on September 10 and within a few days the chassis designs were so far toward completion that arrangements were made to produce two sample trucks. These were completed in an unbelievably short time, and their satisfactory performance began when the paint on the chassis was hardly dry. Driven from Rochester, N. Y., and Lima, Ohio, they arrived in Washington on October 14 and were presented to Gen. Chauncey B. Baker and by him to Secretary of War Newton D. Baker on October 19, which will always be held a red-letter day in the annals of the Society.

The chassis design was based upon the Quartermaster General's office specifications of May, 1917, which were closely followed. The principal changes were these:

	May Spec.	Actual Spec.
Wheelbase limit,	156 in.	160½ in.
Weight limit,	8000 lb.	8800 lb. (aprx.)
Rear tires, diameter,	36 in.	40 in.
Type of tires,	demountable	pressed-on
Radiator surface,		
Frontal,	805 sq. in.	590 sq. in.
Total,	32,220 sq. in.	35,000 sq. in.
Tractive factor (high),	0.0775	0.063
Tractive factor (low),	0.338	0.307
Ignition system,	dual	double
Differential,	locking	free acting
Chassis lubrication,	grease or oil cups	mag. oil feed
Gasoline tank,	30-gal. tank 6-gal. reserve	2 16-gal. tanks
Sprags,		added
Permanent stops for springs		omitted

The following were the guiding factors in designing the Class B chassis; they are set down in the order in which they were given weight. This tabulation was not official but was observed by the author, acting as chief draftsman on this work, and it was generally followed:

(1) Reliability.
(2) Accessibility.
(3) Weight.
(4) Simplicity and ease of manufacture.
(5) Operating economy.
(6) First cost.

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The item of efficiency was more particularly studied and cared for by the parts design groups. Wherever the location or suspension of a part had an important bearing on efficiency it was considered as having greater importance than the second item.

Description of Chassis Units

Possibly as good a way as any to describe the chassis design is to take up the units which enter into its construction.

The frame is of open-hearth steel pressed into shape. The side rails are channel section with parallel flanges which are tapered at the front to reduce weight and give greater room for the radiator. There is a heavy pressed steel cross-member in front of the radiator, one between the front brackets of the rear springs, and one at the rear. A lighter pressed steel cross-member supports the rear of the gear-box, while forged cross-members support the front of the gear-box and the front of the engine. The frame is designed to be as flexible as possible without in any way being flimsy or likely to buckle.

The pressed steel frame allows better distribution of metal than a rolled section, its uniform thickness is of advantage in riveting—there is better assurance of uniformity of metal, and the rounded portion where the flanges meet the web allows large fillets for brackets. The frame is a bit narrower than usual, 34 inches to the outside edges of the side rails, in order to allow a maximum turning angle for the front wheels and to save weight in cross-members and support brackets.

The springs are designed to be nearly flat under rated load. The axles are fastened to the middle of both front and rear springs. This permits the assembling of the front springs without fear of deranging the steering-gear linkage, as might happen if the front axle were not in the middle of the springs and the long end of the spring (on a repair job) were put where the short end should be. The spring seats are also reversible and the steering pivots are given a castor action by inclining the whole front spring. This also enabled us to get a good location for the ball on the axle steering arm and to insure a minimum of stress on the steering-gear and connecting-rod due to axle motion.

The rear springs form the driving connection between the rear axle and the frame, a la Hotchkiss. The dis-

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cussion at Columbus previous to the decision to adopt this type of drive was illuminating, and after all was said the only argument against it was its tendency to throw the rear-wheel brakes out of adjustment if the brake connections were not correctly designed or when the spring under load had a considerable amount of camber. To my mind, however, with Hotchkiss drive, one other point—the slip-joint in the drive-shaft at the forward universal joint—should be given careful attention. Unless the slip-joint is carefully made, thoroughly protected from dirt and properly lubricated, the motion there will result in a serious end-thrust on the universal-joint and the bearings behind it.

By placing the steering-gear behind the rear engine support-arm a long straight connecting-rod was secured. The gear is held by a capped trunnion-bracket and can be readily removed without disturbing the engine or controls. One of the bolts in the gearcase extends through a lug on the bracket and thus relieves the spindle of any bending strains. The column is firmly braced to the dash. The steering-gear case is placed so that the clutch can be removed without disturbing the gearcase. The cap for the gearcase carries a lug that supports the clutch and brake pedals, and makes a very clean bit of design at this point where, on the average truck, there is an entanglement of brackets, shafts and nuts, impossible to adjust without a wrist like a flamingo's neck.

The tires specified were 36 by 5-in. single front and dual rear or 36 by 10-in. single rear. They were to be of the demountable type. After much discussion the change was made to the pressed-on type, saving some 400 pounds in weight and affirming the almost unanimous opinion of the engineers on the job and the tire makers that it was the better type. The evidence from across the water, also, was strongly in favor of the pressed-on type.

The 36-in. rear tires were changed to 40 in., which gave 2 inches more clearance under the rear axle, better adhesion, better rolling on bad ground, lower gasoline consumption, more tire mileage, and an 11 per cent increase in the truck speed for a given engine speed.

The wheelbase, named as 124 in. minimum and 156 in. maximum in the Quartermaster General's office specifications, was increased to 160½ inches. This was deemed advisable in order to keep the proportion of live load on the rear axle below 90 per cent, and at the same time have ample seat-width and a sufficient space between the seat and the steering column.

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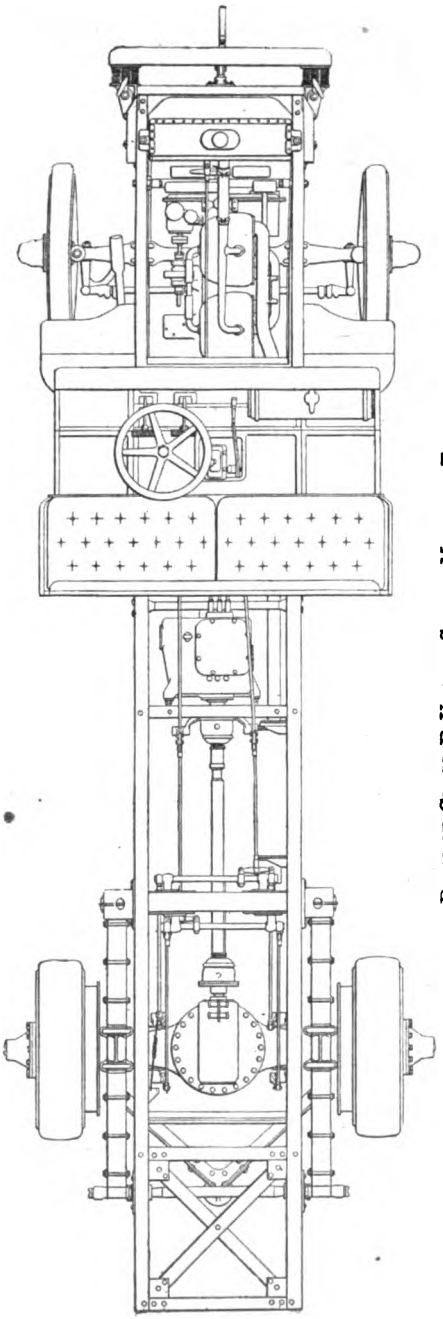

PLAN OF CLASS B UNITED STATES MILITARY TRUCK

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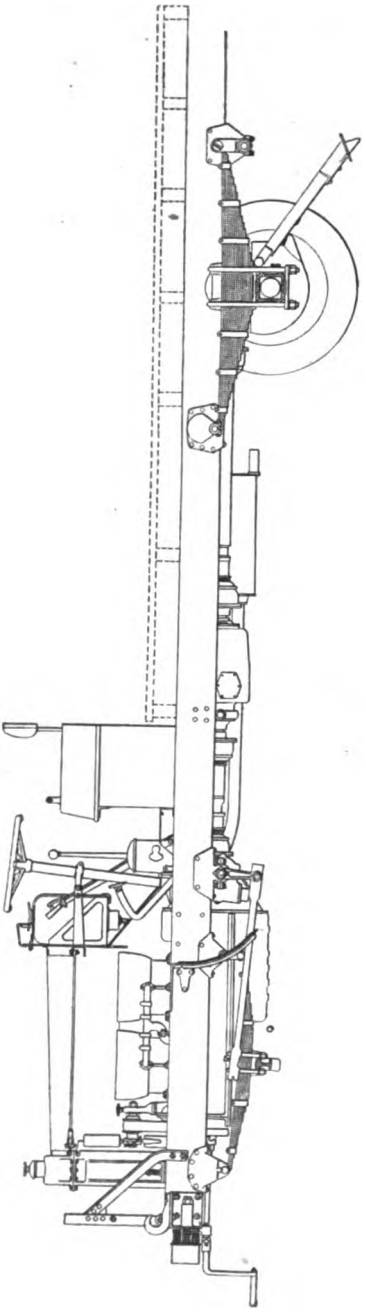

ELEVATION OF CLASS B UNITED STATES MILITARY TRUCK

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Mounting of Units

The mounting of the units in the frame was carefully considered from the following standpoints:

(1) Frame flexure.
(2) Accessibility.
(3) Independent assembling and demounting.
(4) Simplicity.

The radiator has two feet which rest on soft pads on each frame rail. The single holding-down bolt in each foot passes through a clearance hole in the rail and a coil spring between the foot and the nut allows the radiator to take its easiest position when the frame distorts. A rod from the flange of the top tank of the radiator to the dash holds the radiator in an approximately vertical position.

The engine also has a three-point mounting. The two rear lugs on the flywheel housing rest in pressed steel pocket brackets riveted to the frame rails. A vertical bolt passes through each lug bracket and a heavy spiral spring between the bottom of the bracket and the nut. The bracket is so formed that the engine lug will be held in place even if the bolt is lost. It is lined, so as not to chafe the aluminum lug. The third "point" of the engine support is a cylindrical surface on the timing gear cover around the nose of the crankshaft. This swivels in a forged cross-member which is bolted to brackets on the frame rails. This construction is light, substantial and easy to dismount when necessary. An odd feature is that the hose leading from the bottom of the radiator to the pump on the engine is jointed by a piece which is fastened to and passes through this cross-member.

The clutch is housed in with the flywheel and the gearshift is mounted on the clutch housing. The hand-brake hand-lever is mounted on the side of the gearshift housing and is, to my mind, an objectionable feature, as it subjects light aluminum castings and the stud threads in them to the heavy pull in the brake-rod. This pull can be severe when the rear springs flex violently, for the ratchet on the hand-lever is non-yielding and there is bound to be some relative longitudinal motion between the rear axle and the brake rocker-shaft.

The gear-box suspension is three-point. At the rear of the box two forgings bolted to it carry lugs for horizontal transverse pins which are carried by vertical clevis bolts in a pressed steel cross-member of the frame. The third "point" of suspension is in a forging around the nose on the front of the gearcase. This forging is held

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in brackets riveted to the frame. The center-line of the main shaft in the gear-box is a little lower than that of the engine. This gives a slight angularity to the front propeller-shaft and also some motion to the universal-joint trunnions; it thus drops the gear-box cover below the top of the frame and gives a more nearly straight-line drive from the gear-box to the rear axle.

The design of the dash and the gasoline system, which was considerable of a problem, was solved by placing half the supply of gasoline in a tank on the dash and the rest in a reserve tank under the seat. When this was first suggested a hand air-pump was to be provided, as was also a pipe from the reserve tank to the top of the service tank so that the reserve supply could be readily transferred. At present, however, the transfer is accomplished by means of a bucket or can.

Method of Chassis Lubrication

In the chassis lubrication of both Class A and Class B chassis, as originally laid out, it was planned to use

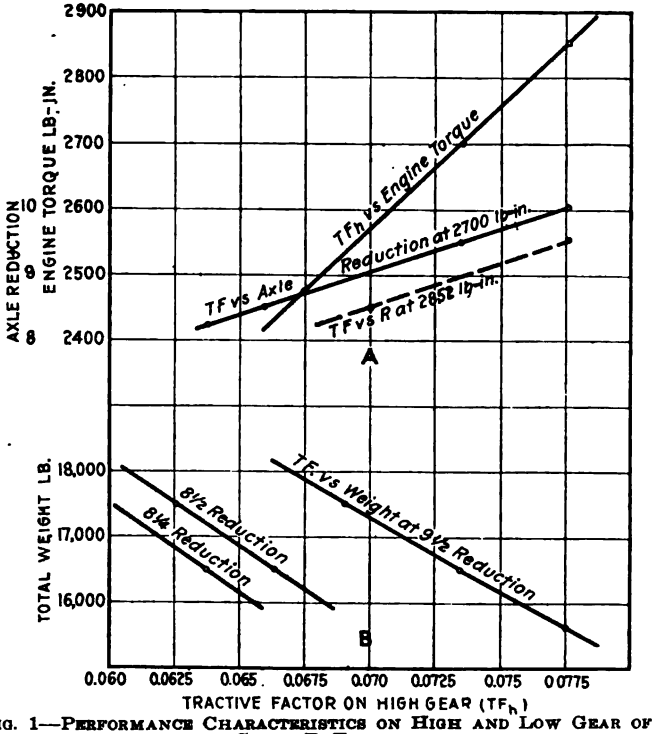

Fig. 1—Performance Characteristics on High and Low Gear of Class B Truck

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oil at almost every point—grease to be used only in the wheel hubs and in the universal-joints. This plan was not carried out on the Class B chassis at some points. All the spring bolts, shackle-pins and brake rocker-shafts are oil-lubricated by the "magazine system." Wicks convey oil by capillary attraction from pockets of considerable size in the bracket castings to the bearing surfaces. The pockets need to be filled but infrequently. Dirt cannot feed to the bearing surfaces even if the oil is dirty. The oil feeds only when the truck is in motion. Seepage from the spring bolts follows down the leaves of the springs and keeps them from rusting. The front axle steering-pins are lubricated in a similar manner and the steering connections are designed for small wick oilers. Grease cups have been put on the connecting-rod to lubricate the

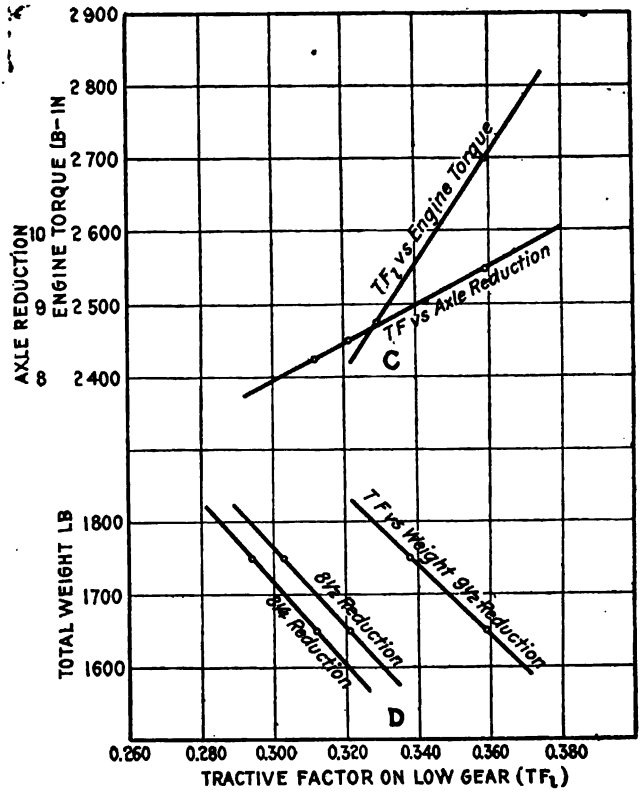

FIG. 2—PERFORMANCE CHARACTERISTICS ON HIGH AND LOW GEAR OF CLASS B TRUCK

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steering balls, on the starting crank, on the brake spindle of the rear axle and at other points.

Paragraph 30 of the Quartermaster General's office specifications lays particular stress on the "tractive factor" of the truck as a measure of its ability to negotiate heavy grades and bad roads. The formula for "tractive factor" is as follows:

$$**T\ F = \frac{8.4\ n\ b^2 s\ R}{D\ W} \times E_m \times E_t$$

n=number of cylinders in engine.
b=cylinder bore, inches.
s=stroke of cylinder, inches.
R=gear reduction between engine and driving wheels.
D=diameter of driving wheels, inches.
W=total weight moved, pounds.
E_m=efficiency of engine compared with A. L. A. M. (now N. A. C. C.) rating as unity.
E_t=efficiency of transmission system from engine to road.

Since $8.4\ n\ b^2\ s\ E_m$ represents engine torque in pound-inches the formula can be written

$$T\ F = \frac{\text{torque} \times R \times E_t}{\frac{1}{2}\ D\ W}$$

Figs. 1 and 2 represent a study of the specifications for the Class B motor truck on the assumption of its coming within the weight limit of 8000 lb. for the chassis. It was predicted that the 4¾ by 6-in. four-cylinder engine would develop not less than 2700 lb-in. average torque between 600 and 1000 r.p.m. The worm-gear reduction is 9.5; one-half the wheel diameter given in the specifications is 18 in.; the transmission efficiency given in the specifications is 85 per cent; the total weight moved was estimated at about 16,500 lb. From these figures (on high gear)

$$T\ F_h = \frac{2700 \times 9.5 \times 0.85}{18 \times 16{,}500} = 0.0734$$

The low gear reduction in the gear-box itself is 5.93 and the specifications call for a transmission efficiency of 70 per cent, whence (for low gear)

$$T\ F_l = \frac{2700 \times 5.93 \times 9.5 \times 0.70}{18 \times 16{,}500} = 0.359$$

**See articles by the author, S. A. E. Transactions 1913, Part I; 1914, Part II; 1916, Part II

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Deducting a road coefficient of 0.0175 as the rolling resistance over hard roads, these figures indicate that the loaded truck should take a 5½ per cent grade on high gear and a 34 per cent on low gear.

Calculation of Tractive Factors

Diagram A in Fig. 1 shows the "tractive factor" in high gear plotted against a variable engine torque and also against different axle reductions. It shows that to get the "tractive factor" called for in the specifications (0.0775) with the reduction of 9.5 adopted, the engine would have to deliver a torque of 2852 lb-in. On test the first engines developed in the neighborhood of 3000 lb-in. torque. It is safe, therefore, to count on 2700 lb-in. as a production standard. The dotted line shows the $T\ F_h$ available at 2852 lb-in., using different gear reductions.

Diagram *B* shows $T\ F_h$ plotted against total weight (W) at different axle reductions (R) and indicates that to get a $T\ F_h$ of 0.0775 the total weight cannot exceed 15,650 lb., which would call for a chassis weight of 7150 lb. Diagrams *C* and *D*, Fig. 2, show similar data for the "tractive factor" on low gear.

Modifying these results to suit the 40-in. driving tires and the 17,300 lb. weight we get for the truck as completed

$$T\ F_h = 0.063$$
$$T\ F_l = 0.307$$

These diagrams show the vast importance of minimum weight as regards truck performance. This is emphasized for the reason that the main criticism of the truck as a whole could be made on the basis of its weight. The primary reason for this weight lies in the tractive factors specified by the Quartermaster General's office and in the gear-box reduction of 5.93. The secondary reason is that the parts design groups were strongly impressed with the arduous service that the truck would be called upon to perform, and each group "played safe" in the matter of the size of parts it recommended.

As a commercial possibility the Class B chassis may be questioned on the score of weight, speed, load rating and gasoline economy. Slight modifications, however, should make it the best 5-ton chassis that rolls. There are some details that are not yet fully developed for the service for which this chassis is intended, but these will undoubtedly be cared for and perfected. There is so much good design in the component parts and in the

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chassis itself that in many ways it will be a criterion for some time to come. An English officer, after having examined the sample trucks and questioned some of our engineers regarding them, summed up the matter well when standing back and viewing one of them critically, he exclaimed, "Well, she looks a job, she looks a job."

DESIGN OF CLASS B TRUCK ENGINE

By A F MILBRATH*

In the early stages of the military-truck design, it was the idea to limit the interchangeability of parts to the complete units, such as engines, axles and transmissions. Several meetings of parts makers were held, with a view to determining the degree to which their regular products could be standardized, so as to make them interchangeable in the chassis.

In the case of the engines, it was found that a standard set of installing dimensions could be determined, which necessitated only slight changes for the various manufacturers. This would have made it possible to replace a damaged unit with one of another make, in case none of the original type was available.

However, the farther this work progressed the more important it was considered by the War Department to have interchangeable not only complete units but the component parts of the units as well. It is obvious that this plan will greatly reduce the number of parts necessary for maintenance of the trucks at the front. When this decision had been made, engineers from the various parts manufacturers were called to Washington to work up suitable designs to meet the requirements of the Government.

TORQUE AND PISTON DISPLACEMENT

The division in charge of the engine design was required to develop an engine with a torque output of at least 2800 lb.-in., which was needed to give the tractive factor predetermined for these trucks. An average of the performance of various engines built in the past showed a torque of about 6¾ lb-in. per cu. in. of piston displacement. Using this figure as a basis, the piston displacement of the Class B engine was determined

*Engineer, Wisconsin Motor Manufacturing Company

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as about 415 cubic inches. Possibly the most important consideration in the designing was that of keeping the wearing parts of these engines very large so that adjustments and replacements of parts would be required but seldom.

Stroke and Bore

It was considered desirable to keep the stroke-bore ratio as high as possible, without obtaining too great a total weight of engine or too great a height overall. The length of stroke was therefore the determining factor in fixing the stroke-bore ratio. A 6-in. stroke was finally adopted. This with a bore of 4¾ in. gives a piston displacement of 425 cubic inches.

Engine Speed and Valve Diameter

The normal speed of these engines was set at 1050 r.p.m., which with the 6-in. stroke gives a piston speed of 1050 ft. per min. The speed of maximum horsepower was also predetermined at about 1500 r.p.m. From past practice it was shown that the maximum horsepower of an engine is developed at a speed at which the theoretical gas velocity through the valves is about 14,000 ft. per min. Assuming an engine speed of 1500 r.p.m., the valve area should therefore be:

$$(4\tfrac{3}{4})^2 - \frac{\pi}{4} \times \frac{1500}{14000} \text{ or } 1.9 \text{ sq. in.}$$

This requires a valve 2⅛-in. diameter in the clear, 2⅜-in. outside diameter, 45-deg. seat, with a lift of $^{11}/_{32}$ in. At the normal engine speed this will give a gas velocity of 160 ft. per sec. through the valve.

Compression Ratio

With the heavier grades of gasoline now in use the compression must be carried somewhat lower than was possible a few years ago; otherwise serious pounding will result in the cylinders. The displacement per cylinder also has some effect on the compression, since a larger cylinder has a smaller wall surface cooling effect per cubic inch of charge than a smaller engine. The past experience of the designers was again called upon, and

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from this a clearance volume of 24½ to 25 per cent of total volume was determined upon. This clearance will give a compression of about 65 lb. gage at slow cranking speeds.

Connecting-Rod and Piston

The connecting-rod of the military engines as compared with regular practice might be considered of the long type, its length being 13¼ in., or 2.21 times that of the stroke. The standard practice varies from about 2 to 2.2 times the stroke. Both upper and lower bearings are laid out central with the rod. The rod is of the I-beam section and the upper end is bronze-bushed for the piston-pin, while the lower end is of the four-bolt type.

The piston is 6⅛ in. long, with three ¼-in. rings of the conventional type, with 45-deg. saw-cut. The piston-pin is 1⅜ in. diameter and is placed 2⅝ in. from the lower end of the piston. A single lock-screw holds the piston-pin in place. This passes through both sides of the piston-pin and piston boss, the lower end of the screw being threaded into the piston boss, while the reduced body size of the screw fits into the upper wall of the piston-pin and boss. The bushing in the upper end of the connecting-rod is 2⅛ in. long, giving an area of 2.92 sq. in. At low engine speeds, when subjected to gas pressures only, this bearing sustains a pressure of 1700 lb. per sq. in. maximum.

At 1500 r.p.m. the greatest pressure due to inertia and gas forces is 1200 lb. per square inch.

Crankshaft Design

As the crankshaft is the backbone of an engine, considerable thought was given to this important part. It is of the conventional three-bearing type. While the governed speed of the engine was to be 1050 r.p.m., it was desired to keep the stresses and bearing pressures low even at 1500 r.p.m. All calculations were therefore made at this higher speed. The stresses considered were three: Those due to gas pressure, to inertia, and to centrifugal forces. Fig. 1 shows these stresses for one position of crankshaft on the power stroke. These forces as affecting the crankpins are given in Fig. 2. The normal gas pressure at beginning of power stroke is assumed to be 280 lb. gage, and the expansion curve to follow the

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equation $p\ v^{1.3}$ = constant. In the calculation of inertia forces the reciprocating weight was taken at 10 lb., including the piston, which weighs 6¼ lb., and the upper end of rod, with wristpin, weighing 3¾ lb. The centrifugal force is set up by the lower end of the rod with bearing, weighing 5½ lb.

In the diagram the forces acting during the suction stroke are represented by dotted lines, those acting during the compression stroke by dash lines, during the power stroke by solid lines, and during the exhaust by dot-and-dash lines. In laying out these forces it was assumed that the gas pressure during suction and exhaust stroke was equal to that of the atmosphere. The lines representing forces during the lower end of the exhaust stroke are not shown, as they practically coincide with the lines representing the compression forces. It will be noted that the greatest pressure exists at the

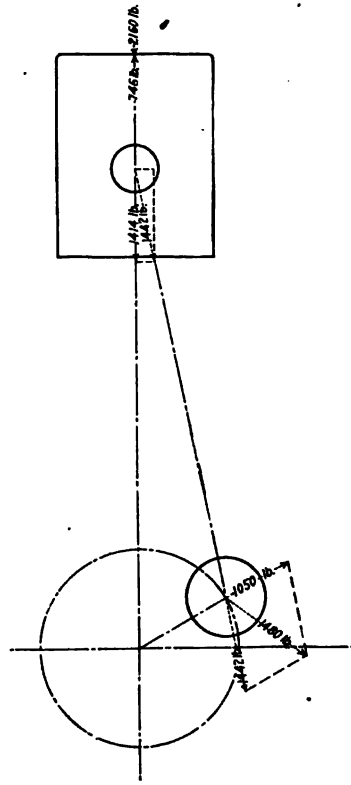

FIG. 1—STRESS ON ONE POSITION OF CRANKSHAFT ON POWER STROKE

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end of the exhaust or beginning of the suction strokes; also that with the exception of the first part of the expansion stroke all pressures are outward from the center of the crankshaft.

The crankpins of the Class B engine are 2⅜-in. diameter and 3 in. long. Deducting for fillets at each end, this gives a projected area of 6.53 sq. in. The maximum pressure on the pins is 520 lb. per sq. in., the minimum 160 lb., and the mean 330 lb. At 1050 r.p.m. the centrifugal and inertia forces are: $(1050/1500)^2$, or 49 per cent of the forces at 1500 r.p.m. At this speed the forces due to gas pressure on the expansion stroke predominate. The maximum crankpin load is then 512 lb. per sq. in., while the minimum and mean are somewhat less than one-half the loads at 1500 r.p.m.

In the calculation of main bearing pressures there must be added to the above forces the centrifugal com-

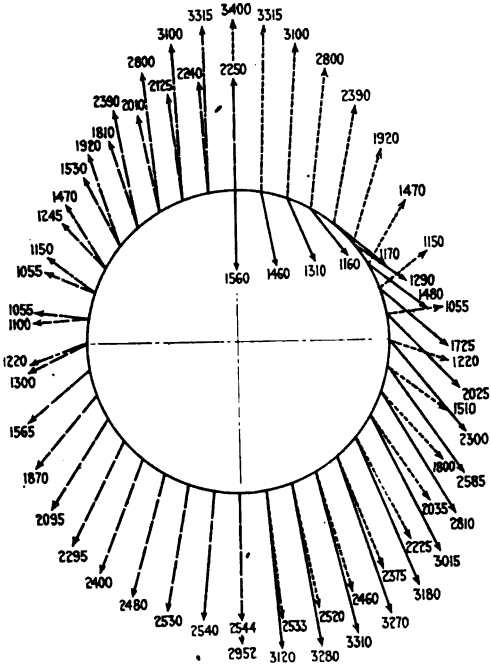

FIG. 2—RESULTANT OF FORCES ACTING ON CRANKPIN

ponent of the short crank-arm and a crankpin. Fig. 3 gives the resultant of the forces acting on the main bearings. These forces are due to gas pressure, inertia

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and centrifugal forces from one cylinder, one set of reciprocating parts, one crankpin and short crank-arm.

Bearing Pressures

In order to decrease the number of spare parts, only two sizes of bearings are used on the crankshaft. The connecting-rod and front main bearings are interchangeable, likewise the center and rear main bearings, which are 2½ in. diameter and 4 in. long; allowing for fillets, this gives a projected area of 9.375 sq. in. The pressures on the front main bearing (in pounds per square inch) are 727 lb. maximum, 32 lb. minimum and 509 lb. mean. The center bearing takes the load from two cranks at the same time. When No. 2 cylinder is on the suction stroke, No. 3 is on the power stroke; when No. 2 is on compression, No. 3 is exhausting, and vice versa. The forces from the two must therefore be combined to obtain the pressure on the center bearing. The result (in pounds per square inch) is a pressure of 903 lb. maximum, 485 lb. minimum and 695 lb. mean.

The rear bearing carries about the same load as the front, plus the weight of the flywheel. The flywheel load is almost negligible, however, as it amounts to less than 15 lb. per sq. in. The maximum pressure on the rear main bearing is 492 lb. per sq. in., the minimum 38 lb. and the mean 355 lb. At 1050 r.p.m. the main bearing pressures are also considerably less than at 1500 r.p.m.

As might be expected, owing to the liberal dimensions of crankpins and main bearings, the fiber stresses in the shaft are low. The crank-arms are of heavy design also, in proportion to the pins, so that the maximum stresses, due to combined torsion and bending, are less than 10,000 lb. per sq. in.

Cylinder Construction

The cylinders are cast in pairs, of the L-head type, with valves on the right-hand side. The heads are removable and held in place by 13½-in. studs. The valve-stems are inclosed by pressed-steel cover-plates. The water-jackets are of very liberal thickness, tapering from ½ in. below to 1¼ in. above. The spark-plugs are screwed directly into the cylinder-heads and are entirely surrounded by water. Cooling water enters the jackets at the lowest point and leaves the heads at the highest point. Thus no drain cocks are necessary, the entire cooling system draining from one point.

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Crankcase and Flywheel

The crankcase is cast of aluminum and is very deep and well ribbed. The parting line between upper and lower half is 3 in. below the center-line of the crankshaft.

The flywheel is inclosed in a No. 3 S. A. E. bell housing. Three-point suspension is employed, two arms cast onto the bell housing and a trunnion bearing on the front gear-cover. Shouldered studs with castle nuts are used for attaching the cylinders. Provision is made for the mounting of electric starters, and generators are mounted as regular equipment. The front gear-cover is of cast iron and is well ribbed on account of its acting as forward support. The lower half of the crankcase forms the oil-pan, and it entirely incloses the oil-pump.

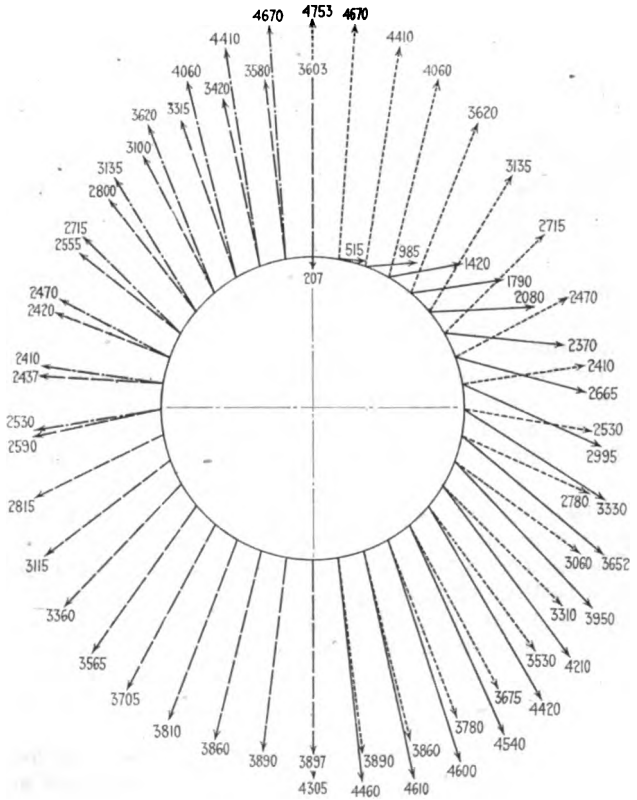

FIG. 3—RESULTANT OF FORCES ACTING ON MAIN BEARING

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Camshaft and Gears

The camshaft is drop forged with integral cams and flange for mounting the driving gear. The cams are $1\frac{3}{8}$-in. diameter, with $\frac{11}{32}$-in. lift. The timing is as follows: Inlet opens 15 deg. past top center and closes 35 deg. past lower center; exhaust opens 45 deg. before lower center and closes 5 deg. past top center. The camshaft bearings are a trifle larger than the cams, so that the shaft can be easily withdrawn.

The timing gears are $1\frac{1}{4}$-in. wide, with 9-pitch teeth cut on an angle of $27\frac{1}{4}$ deg. The drive is from the crankshaft gear through an idler to the camshaft and water-pump gears, and from the camshaft to the generator-drive gear.

The valve tappets are hollow, $1\frac{1}{8}$-in. diameter, of the roller type. They carry adjusting screws 1-in. diameter, also bored hollow.

Oiling System

The oiling system is of the force-feed type, embodying a gear-type circulating-pump. No oil pipes are used, the oil being forced through a passage drilled in the pump body to a header extending the full length of the crankcase, then through drilled passages to the main bearings. Grooves are provided in the main bearings, which are in constant communication with the oil-holes drilled through the crankshaft to the connecting-rods. Oil tubes on the connecting-rods lead oil to the piston-pins. A relief valve set for 10-lb. pressure is fitted to the front end of the main oil header, and the overflow from this valve lubricates the gears.

The oil-pan or lower half of the crankcase is so formed that the oil will drain toward the center of the sump, even though the engine is inclined, as in climbing or descending a hill.

The oil drains from the crank chamber through a spout into a settling chamber, where any water or mud settles, and can be drained off from time to time. As the oil rises in the settling chamber it finally overflows the top into a large strainer, which completely surrounds the settling chamber. Through this strainer the oil flows by gravity into the sump proper.

The oil-pump is supported from the upper crankcase and extends down into the sump. Another strainer surrounds the oil-pump; hence the oil is thoroughly cleaned.

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No parts whatever outside of the settling chamber and large strainer are attached to the lower half of the crankcase, so this can be dropped by taking out the bolts, without interfering with anything else. No oil float of any kind is used, but a graduated test rod is fitted, which can be withdrawn to ascertain the amount of oil in the engine.

Methods of Ignition

Two entirely independent systems of ignition are incorporated; a high-tension magneto, which is driven from the rear end of the water-pump shaft, and a combined timer and distributor with coil mounted at the forward end of the engine, on the left-hand side, and driven through helical gears from the water-pump drive-shaft. Two sets of spark-plugs are fitted into the heads of the cylinders, and the engine can be run on either one or both of the systems.

Inlet and Exhaust Manifolds

The inlet and exhaust manifolds are on the right-hand side of the engine. A heating chamber is cast integral with the inlet manifold near the center and where the vertical branch from the carbureter joins the horizontal section. This heating chamber bolts directly to the exhaust manifold at a point where an opening is cast in the latter, so that the hot gases can circulate around the inlet manifold. This forms a hot spot about 4 or 5 in. long, which helps to vaporize the fuel and permits the use of low grades of gasoline.

Fan for Cooling System

A 22-in. fan, especially designed for these engines after a long series of tests, is mounted on a rigid bracket, bolted to the crankcase. The fan is provided with a vertical adjustment for taking up the slack in the belt. This, however, will seldom be necessary, as a 2-in. belt and large pulleys are fitted, which will drive the fan when set up with only a slight tension.

Engine Governor

An entirely inclosed governor, driven from the forward end of the generator shaft, is built into the engine. The centrifugal members of this governor consist of four

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1-in. steel balls, held in place by a spider. The governor revolves at about one and one-half times engine speed. The thrust from the balls is 34 lb.

On their outward movement, these balls travel against the face of a 45-deg. disk, which also gives the balls a longitudinal movement against a collar free to slide on the governor shaft. The collar has a movement of 3/16-in., which is transmitted through a vertical lever, multiplying the movement threefold. A link with ball joints connects the upper end of the lever with a throttle spindle in the inlet manifold. A spring is fitted to the vertical lever, and an adjustment with a seal is provided.

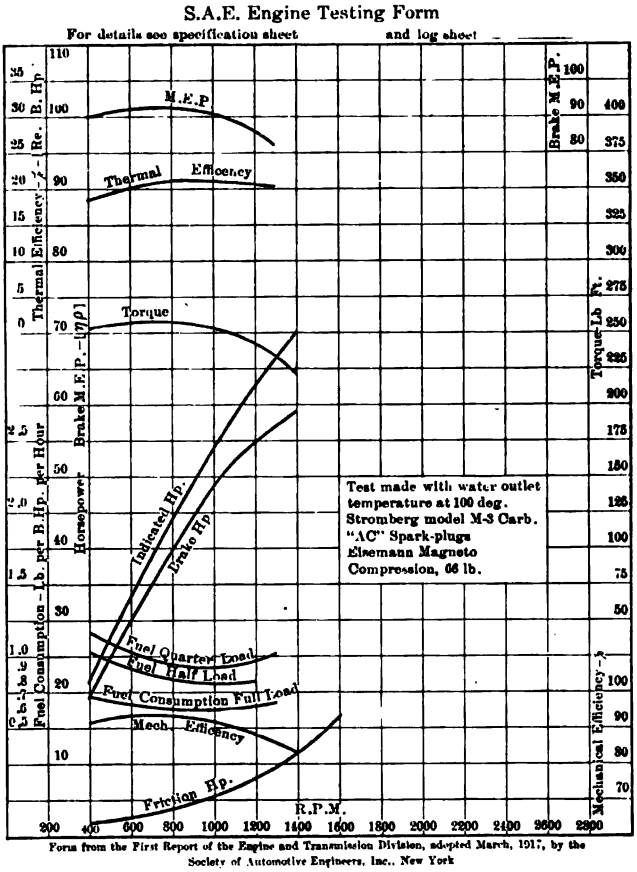

FIG. 4—PERFORMANCE CURVES FOR CLASS B TRUCK ENGINE

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Performance Curves

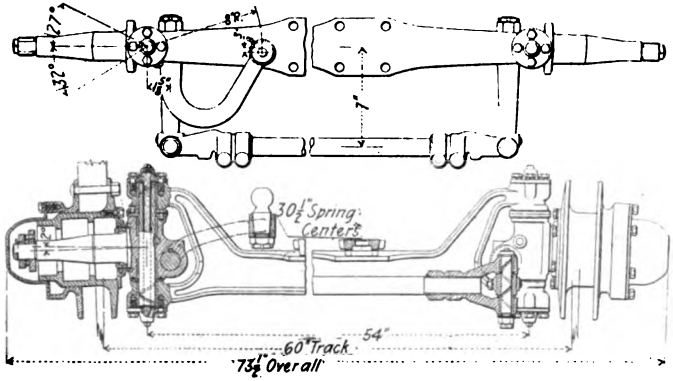
Fig. 4 gives the performance curves of one of the first engines built. The maximum torque is developed at 800 r.p.m., and is equal to 258 lb-ft., or 3096 lb-in.

With a piston displacement of 425 cu. in. this gives a torque of 3096/425, or 7.27 lb-in. per cu. in. piston displacement. Fuel consumption is given for quarter, half and full load. At 1550 r.p.m. the maximum horsepower would be developed.

DESIGN OF MILITARY TRUCK AXLES

By G W CARLSON*

When the Government first summoned engineers to Washington to design a U. S. army truck, the axle engineers were divided into three groups, representing worm, internal gear and double reduction types of final drive. The program was to design and build two or more of each type. These were then to be tested, and the selection was to be based upon all-around performance as shown by the tests.



FRONT AXLE DESIGNED FOR CLASS A MILITARY MOTOR TRUCK

The limited time however upset this procedure, and it became necessary to make a quick decision, at least in the case of the Class B design. It is not my purpose to discuss the purpose the War Department had in view in its selection of worm type gearing for the Class B axle. Suffice it to say that the wide use of this type as well as

*Engineer, Timken-Detroit Axle Company

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production conditions were large factors. Its performance and absolute reliability through critical periods eliminated all inclination on the part of the engineers to introduce features of an experimental nature in the truck. The governing keynote is sturdiness, simplicity of construction and simplification of the field service problem.

An instance of the last is manifested in the Class B axles particularly, in which only three bearing sizes are used in the front and rear—the front inner bearing interchanging with the worm bearings—the rear wheel bearings being alike and interchanging with the differential bearings, and the front wheel outer bearing being the only odd size. The front carries only one size of screw, a total of forty-four being used. Such is also the case in the rear, where the only right and left-hand parts are the wheel-spindle nuts and the brake anchors. The brake levers, brake shafts, shaft brackets and toggles are all interchangeable.

CLASS A SAMPLE MILITARY TRUCK

CONSTRUCTION OF FRONT AXLES

The Class A and B front axles are, except for proportionate dimensions, the same. The hub closure is designed with the object of presenting the smallest diameter for the felt rubbing face. A flange on the felt retainer rotating within a groove in the knuckle prevents the dirt from reaching the felt, and an additional groove immediately outside the felt is intended to keep any water from reaching it, except possibly at the bottom.

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Particular attention has been paid to the lubrication of the knuckle-pin. This is made of carbon steel, hardened and ground, and the bushings are of similar material and pressed into the ends of the center forging, this construction providing a wide bearing spread. The hollow pin contains an oil wick, the ample dimensions of which allow for a generous supply of oil.

Since the ordinary grease commonly used for lubricating this particular place leaves much to be desired, it was argued that even at the expense of considerable leakage oil would be preferable. The experience with the samples is ample testimony of the soundness of this reasoning.

CLASS A SAMPLE MILITARY TRUCK

The thrust washers are grooved, the oil entering through recesses next to the pin. The dust device consists of a brass ring floating in opposite circular grooves in the washers. The bottom of the knuckle is closed by means of a similar expanding ring. These rings are intended more to keep out the dirt than to keep in the oil.

The cross tube end-pins are fastened to the arms in

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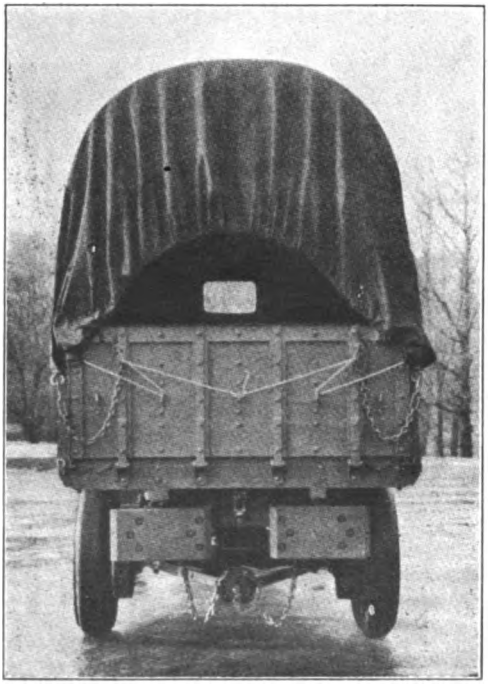
order to allow for the maximum bearing surface with minimum weight. A draw key is used for this purpose. The bushings are pressed into the end forging. The lubrication is by oil. No attempt is made to hold the oil, a self-feeding oiler being used.

The regular Elliott type was chosen because of the opportunity it presented of bringing the pivot centers close to gage line, and because it was a simpler manufacturing proposition.

The center forging material is No. 1035 S. A. E. steel, heat-treated; the knuckles and arms No. 3130 S. A. E. steel, heat-treated; hubs and flanges, malleable iron. In short, the S. A. E. materials and heat-treatments are used wherever possible.

The steering-arms are forged to shape, except for a hand-bending operation on the ball arm. They are secured in the knuckles by taper fits of generous dimensions, and keyed.

The Class AA front axle is of similar type, without



CLASS A SAMPLE MILITARY TRUCK

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the refinements of detail used in the A and B designs. This was necessary because of the limited weight allowed for the lightest axle. The general construction is distinctly of the passenger type. The lubrication is all with grease, the limitations imposed, as before stated, not permitting a design suitable for oil without approaching features of an experimental nature.

The hubs, unlike those of the two heavier types, are of pressed steel, which embodies the necessary strength with additional lightness. The material is otherwise the same as for the A and B, part for part.

Class B Rear Axle

During the early stage of development of the Class B truck, it became apparent that the weight would far exceed that of the commercial 3½-ton truck. This, coupled with the severity of service, brought about a design which, for strength, compares favorably with the average 5-ton commercial rear axle. The weight alone, however, was not the only determining factor. The War Department specification for the locking differential necessitated a shaft diameter of a size that put the bearings and tubes into the 5-ton class. It was not considered safe to calculate the shaft stresses except by the engine torque. The material selected for the shaft was a steel that would present the least uncertainty in heat-treatment.

These conditions made it possible, without any sacrifice, to use one size of bearing for the wheels and differential. The drive-plates are shrunk to the shafts (ten splines being used in each end) and piloted into the hub flange, to which they are secured by fourteen ½-in. bolts. Studs were considered, but through-bolts were given the preference.

The housing construction is of a Timken type, either cast or pressed steel, with a square section from the center bowl out; the tubes are pressed into place, a retaining screw being provided for safety. The tube extends to the bowl of the housing, and a reinforcing plate is fitted over the inner end and riveted or welded to the top and bottom of the housing shell.

The forged steel spring-seats are in two parts—top and bottom—each fitted over the housing on machined surfaces and doweled for side position.

The worm and gear are of the David Brown type, with an 8⅝-in. center distance. The pressure angle is 30 deg., the linear pitch 1.1562 in., and the lead angle is

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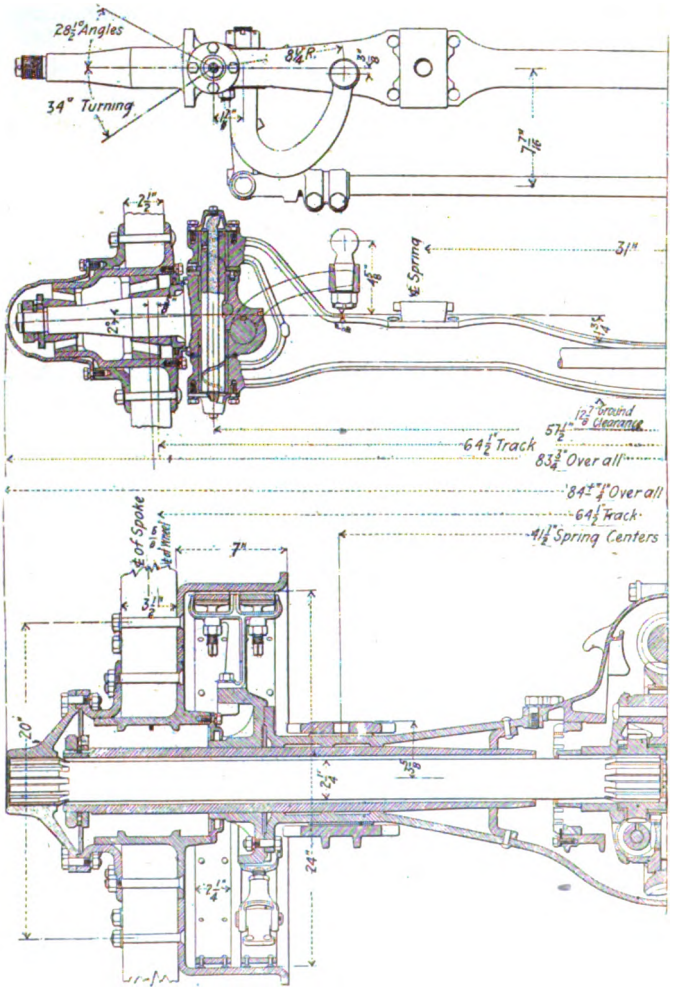
24 deg. 20 min. Compared with the sizes used in commercial trucks, this design appears somewhat inadequate. The limiting factor, however, is ground clearance, which



PLANS AND SECTIONAL VIEWS OF FRONT AND REAR AXLES

the military authorities considered so necessary as to sacrifice possibly the maximum life of the worm gear.

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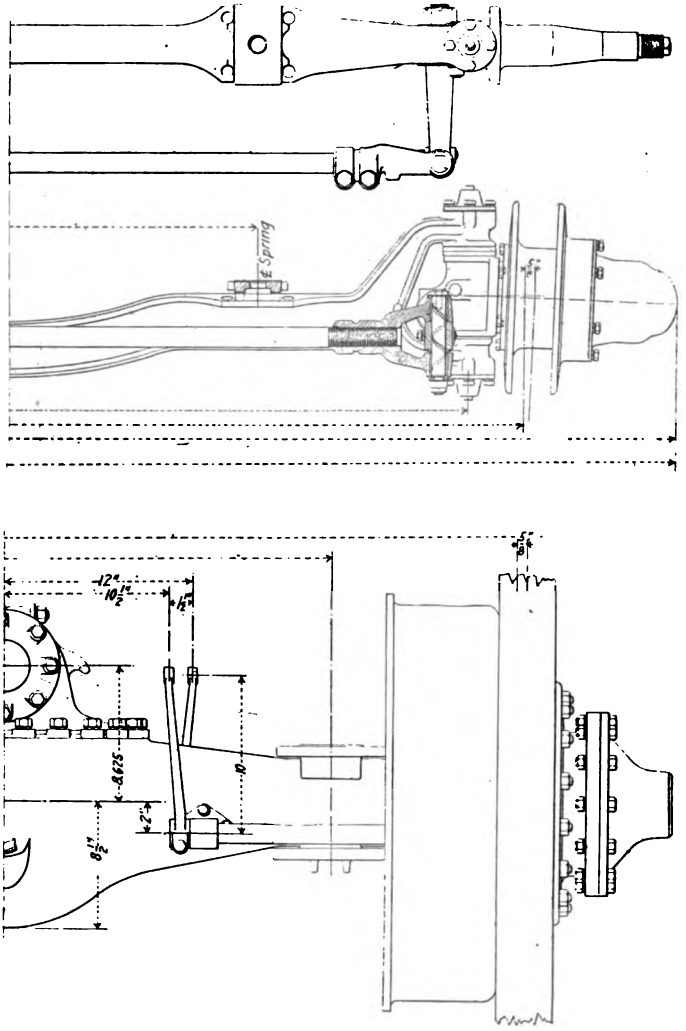
It might be mentioned that the samples, giving due consideration to the comparatively short mileage, do not indicate that such a sacrifice has been made.



DESIGNED FOR THE CLASS B MILITARY TRUCK

The gear is mounted on the differential by means of corrugations 3.16 by 1/16 in. on one side for one-half the

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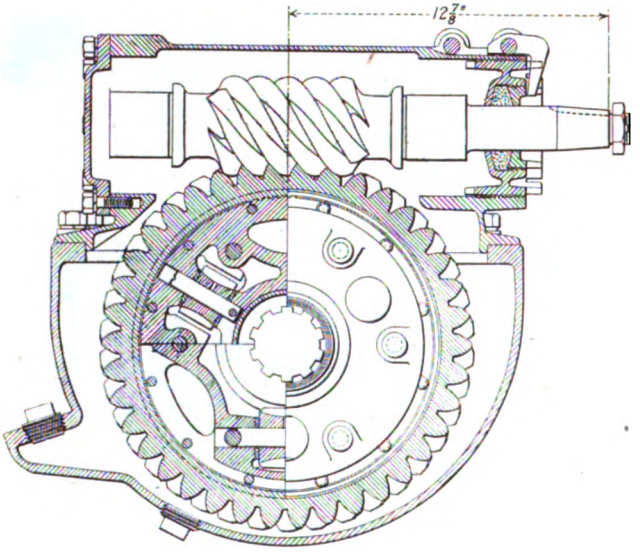
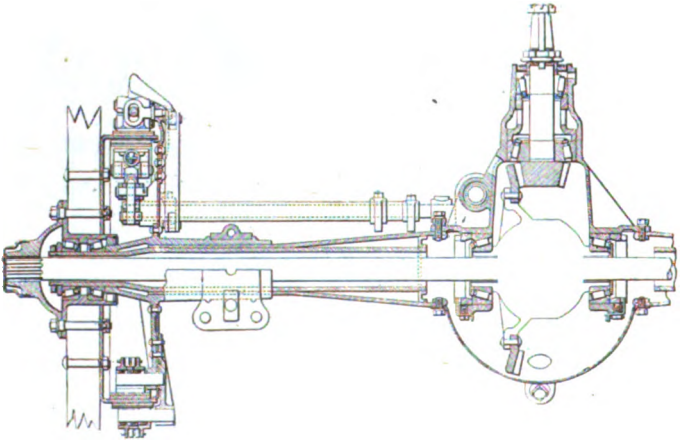
width of the gear seat and piloted by the other side. It is secured against side-thrust by a ring held by twelve rivets and against a shoulder on the opposite side. This



CROSS-SECTION SHOWING CLASS B WORM GEAR

mounting, although not new, gives large space for the differential, as well as economy in bronze.



CROSS-SECTIONAL VIEW OF REAR AXLE FOR CLASS AA MILITARY MOTOR TRUCK

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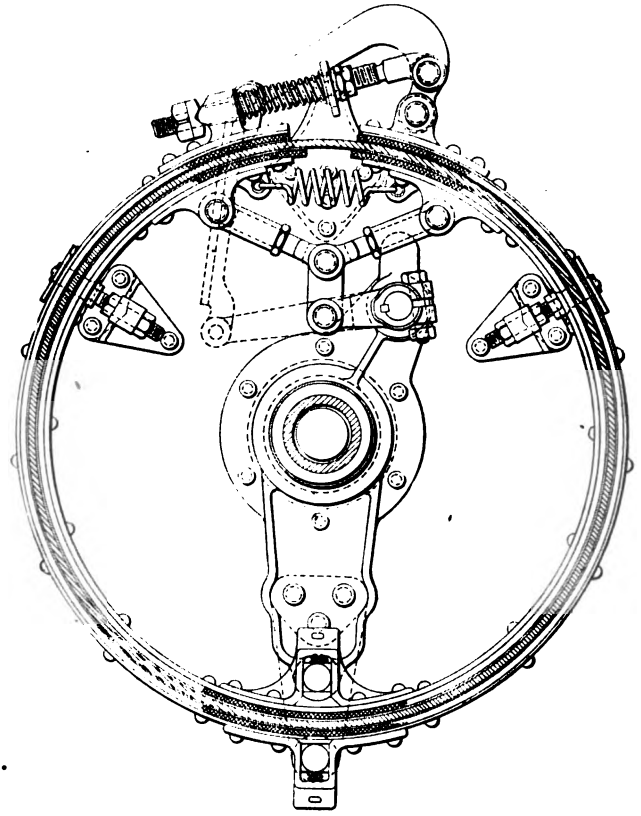
Differential Possibilities

The differential originally chosen was of a well-known locking type, but because of the comparatively small number in use for this capacity a common type was substituted. The locking differentials on the sample trucks have thus far given complete satisfaction and seem to be



BRAKE SYSTEM USED ON CLASS B REAR AXLE

adequate from the standpoint both of strength and wear.

The differential worm and gear are mounted on a carrier piloted at the top of the housing. The gear adjustment is made by slotted and threaded rings in back of the bearing cups. The cups are clamped to the main pedestals by caps, nickel steel studs 1-in. diameter being

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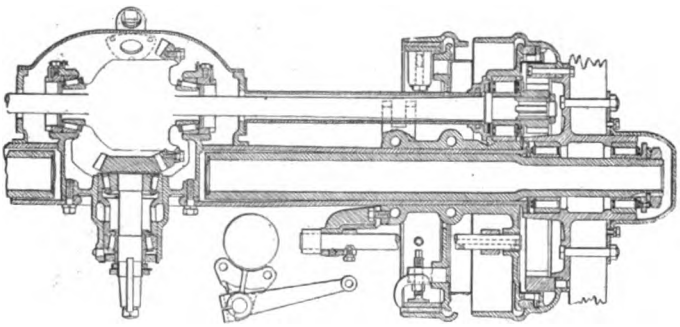
used. The whole unit can be removed from the housing without disturbing any adjustments.

Throughout the axle the various parts have, wherever possible, been so designed as to permit assembling and



CROSS-SECTIONAL VIEW OF REAR AXLE FOR CLASS A MILITARY MOTOR TRUCK

disassembling as the following units: (1) The shaft and drive-plates; (2) the wheel, drum, bearings and dust washer; (3) the brake anchor and bands, levers and shafts; and (4) carrier and parts.

The brakes are side by side and are made of rolled stock, toggle operated, 24-in. diameter and 2½-in. wide. The proper place for brakes is unquestionably on the rear wheels, provided they can be made of a size large enough to be dependable. Since the War Department permitted a 24-in. diameter the problem was merely one of designing a brake easy of adjustment and of disassembling or relining. The toggle brake in this instance was given preference because it made accurate forming of bands and machining unnecessary. The objection raised on account of the rattle is of minor importance when the braking power available is considered.

Besides the anchor and supporting pin (which is square) three additional adjusting screws are provided for each band, designed so as to make adjustments possible without removing the wheel and drum. These have mushroom heads to prevent pounding into the softer metal of the band. The drums are pressed steel, flanged to give rigidity. S. A. E. materials and standards are used throughout.

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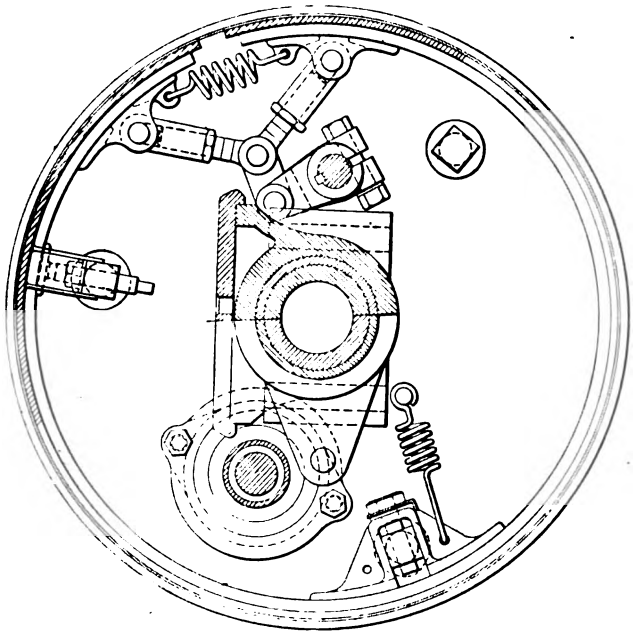
All bearings are tapered roller except the internal gear spur pinion and the wheel bearings on the Class A truck, which are of straight roller type. The manufacturers making the internal gear axles recommend this practice on the ground that failure to adjust would cause a misalignment of the gears.

Oilers and grease cups have been omitted wherever possible and pipe plugs substituted. All nuts of large diameter and fine threads are case-hardened.



VIEW OF CLASS A REAR BRAKES

CLASSES A AND AA REAR AXLES

The internal-gear type axle was selected for the Class A rear axle; first, because of the available manufacturing facilities, and second, because of the high ground clearance desired.

The countershaft was placed in the rear of the load carrying member; first, it makes possible a simple assembling job; second, gear adjustment can be made by simply

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removing the rear cover-plate (as in current passenger practice); and, third, lighter weight is thus secured.

The question of the loads added to the wheel bearings with this construction was considered, but was found to be negligible as contrasted with those due to the road shocks.

The load-carrying member is a pressed-steel housing. The differential carrier is bolted to the inner face of the front flange. The drive-shaft tubes are ⅛-in. wall tubing, flanged at the inner end and bolted to the aluminum case, and piloted into a felt ring at the outer end.

The bevel gears are somewhat larger than necessary owing to the fact that provision had to be made for a locking differential. The wheel brake (emergency) is of the toggle type, the service brake being attached to the transmission.

The Class AA rear axle is of the generally accepted full-floating passenger-car type, selected because of the large production possible. The ring gear is 12⅞-in. diameter, 61-tooth pinion and 11-tooth, 4¾-in. pitch spiral-bevel gear. Current practice has been followed in each feature.

TRANSMISSIONS FOR THE CLASS B TRUCK

By A W COPLAND*

The first meeting of the newly formed Transmission Division of the S. A. E. Standards Committee was held May 2 and 3, 1917, in Cleveland. Some two weeks before this time the Motor Transport Board of the War Department had issued standard specifications for Class A and Class B gasoline motor trucks in the preparation of which the Truck Division of the S. A. E. Standards Committee assisted.

The first business taken up by the new Division was a study of the standard specifications as they directly or indirectly influenced the transmission and clutch design. Some minor changes in the specifications relating to transmissions and clutches were suggested. These were accepted by the Government and incorporated in the later edition of the specifications.

At this time the plan was to accept any commercial engine, transmission and axle that would come within the specifications and use them in trucks that would

*President, Detroit Gear & Machine Company

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measure up to the requirements. However, the design of the fastening points and connections for each major unit was to be standardized so that each unit would be interchangeable with any other unit of its class. The Transmission Division took up this work with great earnestness and held several meetings during May and June.

Nearly every transmission maker in the country submitted at least one design and some manufacturers presented a new design at every meeting. It was found necessary to make changes in the fastening points of every transmission then in production which was submitted for consideration; in most cases the gears and countershaft were the only parts not changed. One maker insisted on having the size of holes in the rear supporting-arms of his transmission made the standard, declaring it was the only thing on the outside of his design that had been left to him. Real constructive work was accomplished by the free discussion and friendly criticism of each other's designs.

Engineering Discussion

The following incident will illustrate the practical value of these frank discussions: Mr. A. remarks: We have never had an entirely satisfactory thrust bearing and throw-out device for our clutches; we attempt to lubricate the thrust bearing by delivering oil from the transmission through a passage in the shank of the drive-gear. This passage was connected by radial holes with the thrust bearing. If we make these holes large enough to insure sufficient lubrication at all times, occasionally the flow will be so excessive as to drain the transmission. We have tried a felt damper in the drive-gear but this has proved only partly successful because so many different kinds of transmission lubricant are used. Now, I have noticed that Mr. B. does not provide any means for lubricating the thrust bearing, but uses a type of thrust bearing quite different from ours. As he has been using this bearing for a long time we have decided to adopt Mr. B.'s type of thrust bearing.

Mr. B. squirms a bit and then confesses he has not been entirely free from thrust-bearing trouble and has even thought seriously of trying the type Mr. A. was about to abandon. Everyone was thus brought into the discussion, and it was decided to make a careful study of the problems involved in the design of the clutch throw-out. This was done, and eventually led to a new arrangement of the parts which insures uniform distribu-

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tion of pressure on the bearing and permits lubrication from the outside with a minimum of trouble.

A gratifying amount of constructive work was brought about by these confessions. Patriotism impelled each one to disclose everything he knew about transmission design and to state freely his experience with the devices suggested. Those with improvements which they expected to incorporate in their product sometime in the future submitted ideas with commendable frankness. With nearly all the transmission makers of the country working together in such a splendid spirit, great progress was made, and at a meeting held June 2 in Washington, four makers decided to join in designing a new transmission for the Class A truck. Several other makers signified their willingness to adopt the new design later if they were called upon by the Government to make transmissions for Class A trucks.

As so much preliminary work had been done in previous meetings and by the engineering departments of transmission makers, the main features of the new transmission design were quickly decided on, and the Ball and Roller Bearings Division of the S. A. E. Standards Committee was requested to recommend bearing sizes. The sizes of ball bearings recommended by it were considerably larger than those used in commercial practice, but the bearing engineers insisted that for military use they could not approve smaller sizes. Stress diagrams were made for all bearings, gears and shafts. At the meeting held July 20 in Columbus a tentative design was submitted. At this time drawings of most of the details had been made, but at this historical Columbus meeting the Government told us it would adopt one design only for each detail part of the A and B trucks "to the last screw, bolt and nut." The transmission makers had progressed so far in this direction that it was an unbroken continuance of their work to consider the transmission and clutch for the Class B truck, which, the Government stated, was needed first.

Details of Transmission

The Government asked for volunteers among S. A. E. members to go to Washington and design two sizes of military trucks. The desired number quickly volunteered and were called to Washington July 30, when the different groups commenced at once to consider their particular problems.

The B transmission is of the amidship type with

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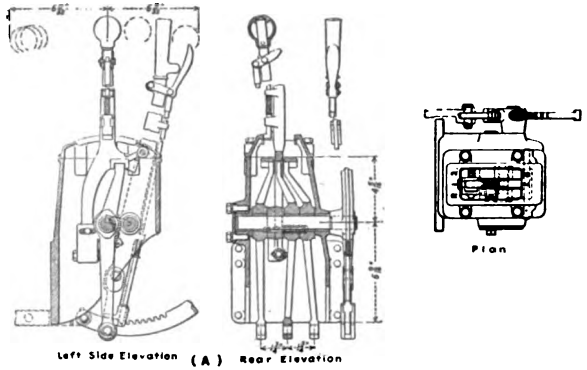

DETAILS OF TRANSMISSION AND CONTROL FOR CLASS B TRUCK

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four speeds forward and one reverse. The main shaft and countershaft are placed in a horizontal plane with the idler gear-shaft above and to the left, looking from the rear forward. The horizontal type was adopted in order to get the ground clearance specified, 18 in. below the transmission at a point midway between the front and rear wheels.

The transmission weighs about 230 lb. The case is of aluminum. The front is supported by a 6-in. trunnion within which is mounted the outer drive-gear bearing. The rear is supported by two steel forgings bolted and doweled to the back of the case. The fact that the mainshaft and countershaft are in a horizontal plane, and that the idler gear is placed above minimizes the depth below the mainshaft.

The gears are 5-7 pitch, 6 in. from center to center of main shaft and countershaft. The width of gears increases from 1¼ to 1½ in. from the constant mesh gears to the low, according to the tooth pressure developed and the work it is expected each speed will be called upon to do.

Values of Speed Ratios

The speed ratios are 5.93 to 1 on first and reverse speeds, 3.23 to 1 on second, 1.76 to 1 on third, and direct on fourth speed. As the rear axle reduction is 9.5 to 1 this gives a total reduction of 56.335 on first and reverse.

It is common practice in transmission design to have the reduction considerably greater on reverse than on first. The ordinary four-speed transmission arrangement with a ratio of 5.93 on first would have one of 6.78 on reverse. A total reduction of 56.335 was considered ample to secure the tractive factor desired in the Class B truck under almost any road conditions, so it was decided to make the ratio on reverse speed in the transmission the same as on low; all working parts to the rear of the transmission can thus be made lighter than would have been required if the usual practice had been carried out of having the reduction reverse greater than on low speed. The countershaft first-speed gear is forged integral with the countershaft. The other countershaft gears are held with a single key.

The reverse idler gears are made from an integral forging. The shifter slot is between the gears. All gears are made from forgings. The mainshaft is 2¼-in. S. A. E. six-spline, ground all over except where threaded. A clearance is provided over the splines in the keyways of gears, the working fit being between the

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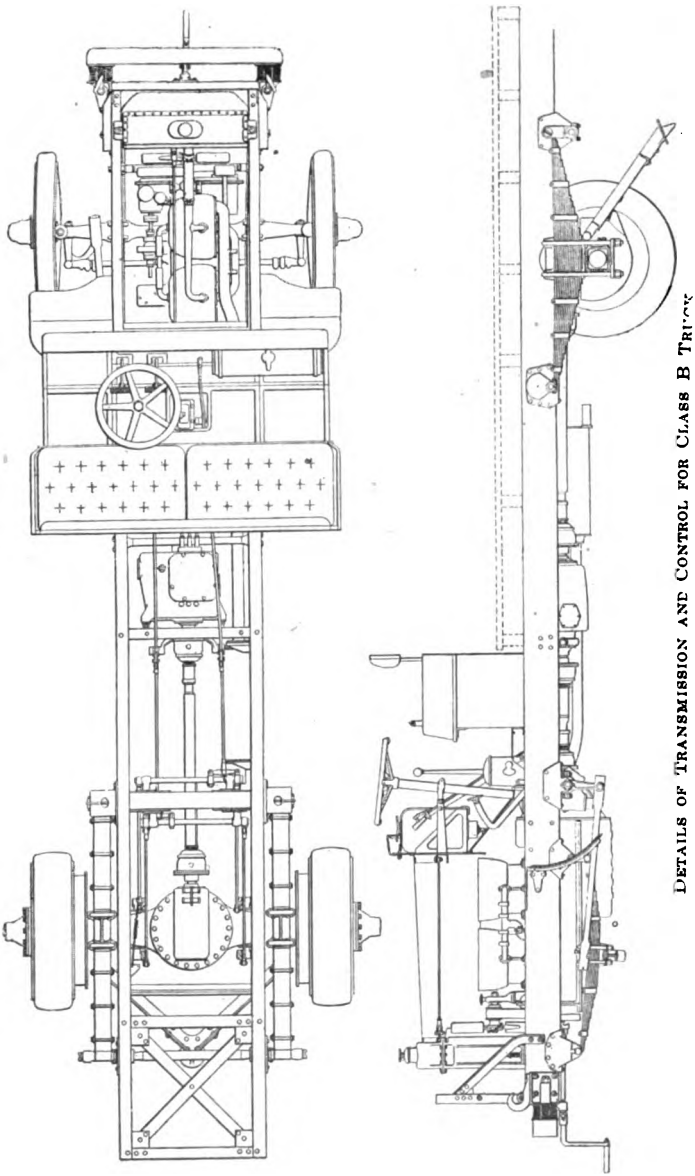

DETAILS OF TRANSMISSION AND CONTROL FOR CLASS B TRUCK

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ground bore of the gear and the small diameter of shaft between the splines.

All shafts are specified alloy steel equal to 3½ per cent nickel in physical properties, properly case-hardened. Gears may be made of alloy steel equal in physical properties to 3½ per cent nickel. The steel may be high carbon properly heat-treated or low carbon properly case-hardened. All the gears are now being made of 3½ per cent nickel steel, low carbon case-hardened.

The lower part of the mesh lock-plungers are wedge shape. Instead of leaving a thin straight end, the end is machined concave and rounded off. This design gives greater wearing surface than the cone-pointed plunger and the concave end prevents the plunger from turning around. There is no interlock in the transmission but a positive interlock is provided in the control set. The control set is a rocking-lever type.

Annular ball bearings are used for drive-gear shank, mainshaft case bearing and countershaft. The mainshaft spigot bearing is of the solid roller type running directly on the pilot on the end of the mainshaft and in the end of the drive gear. Plain phosphor bronze bearings are used for the idler gears. The drive-gear shank terminates in a 1½-in. S. A. E. taper and the mainshaft in a 1¾-in. S. A. E. taper.

Current Practice Followed

Although the several units of the Class B truck were designed by separate groups, the different groups kept in close touch with each other and the result is a truck of balanced design. The Class B transmission is in no sense a compromise design. Each part of the design had practically the unanimous approval of the transmission engineers on the job, and in most cases represents successful current practice. It is confidently expected to give a good account of itself under the most trying conditions on the battle front. Throughout the whole design S. A. E. Standards were used just as fully as they were available.

The speed-change H-plate is the S. A. E. recommended practice. No rocker-arms are used in the transmission. The shifter-forks are fixed to the shifter-rods by hardened screws having special taper ends. The shifter-forks are as short as the diameter of gears will allow. The arrangement is such that the shifter-rod nearest to the idler gear is the one used for reverse. Grooves of 90-deg. V-shape are used for the shifter-rod notches.

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FUEL FOR AUTOMOTIVE APPARATUS

By Dr E W Dean*

The present paper deals with certain problems common to the two industries of automotive engineering and petroleum refining. A brief preliminary discussion seems to be in order to explain why a chemist is elected to present information apparently not related to his normal field of work.

The Bureau of Mines has for several years been conducting scientific investigations in the field of petroleum technology. This work was begun through the establishment of a chemical laboratory for the testing of oils. It was, however, recognized at an early date that little of practical importance could be accomplished if activities were limited to such a narrow scope. When the present petroleum division of the Bureau was organized the chemical section of this division was made to include with its laboratory investigations studies in the fields of petroleum refining and utilization.

The Bureau was early impressed with the general failure of producers and users of petroleum products to work together in dealing with the many problems that are of common interest. It is now thought that no more important end can be accomplished than to interest refiners and automotive engineers in the advantages of cooperation.

The principal petroleum products in which automotive engineers are interested are gasoline and lubricating oil. At present the latter does not seem to demand as much attention as the former and although there is little doubt as to the advantage to automotive engineers of a knowledge of the manufacture and properties of lubricants, it seems most desirable at present to concentrate efforts on obtaining a solution of the fuel problem. Fuels, as referred to in this paper, will be understood to mean the liquid fuels employed for operating internal-combustion engines.

Gasoline Problem of Chief Interest

During the past few years the demand for gasoline has increased far more rapidly than the supply of raw material from which it is produced. Consequently there has been a general change in refinery practice with the production of so-called heavy gasolines, which are actually of relatively low average volatility. Although this

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change in quality is not necessarily deterioration, it has brought to the attention of designers and users of internal-combustion engines problems that did not exist in the past. Coincident with the change in quality there has been an increase in market price and a threatened condition of shortage in supply, so that the automotive engineering industry now has for solution the complex problem of devising equipment for the utilization of fuel of low volatility and at the same time getting a greater production of effective power from the quantity of gasoline used.

Properties and Production of Gasoline

Before beginning the most important part of this paper—that which discusses the possibility of increasing the supply of fuel and the efficiency of its utilization—it is desirable to consider briefly the characteristic qualities of gasoline and the possibility of variation among these as influenced by refining methods. It is already understood by automotive engineers that the name gasoline is decidedly indefinite in its meaning and that it includes a heterogeneous class of petroleum products that have in common only the power of enabling automobilists to start and operate their engines. The following discussion is intended to show some of the characteristic differences among types and grades of gasoline and to explain and perhaps justify the changes in quality that have occurred recently.

Review of Refinery Methods

There are now three general methods by which gasoline is obtained from raw material. The oldest and still the most important is the ordinary refinery process of distillation from crude petroleum. A second method, also used in refineries, is the pressure-cracking process, in which, instead of crude petroleum, heavy distillates, such as gas oil or fuel oil, are used as raw material. The third method involves the extraction of gasoline from natural gas, either by the absorption or the compression process. The product thus obtained is commonly called casinghead gasoline.

An adequate discussion of the details of any one of these processes is beyond the scope of the present paper and it is therefore necessary to pass this interesting field with a brief word of mention. It may be stated, however, that the straight distillation method is highly developed and that improvements in it offer only moderate possibility of increasing the supply of fuel. Some gasoline and naphtha still go into the kerosene fractions, but wastes

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of this sort are not important in the present situation. The cracking process is as yet only partly developed, and in the future its use will undoubtedly be considerably increased. It is not to be expected, however, that there will be more than gradual development in this field even if the Standard Oil Company should decide to license the Burton process for general use at an early date.

Other cracking processes are being installed on a commercial scale but as yet these seem to be limited to the small unit stage. The extraction of gasoline from natural gas is likewise a method that is growing in use and it is difficult to predict just what can be expected from it on the basis of data now at hand. The recently developed absorption process makes it profitable to extract gasoline from gas not sufficiently rich to be treated by the compression process, but resources in this direction probably cannot be expected to develop in more than gradual fashion.

Properties of Straight Refinery Gasoline

The most familiar type of fuel, that produced from crude oil by the distillation method, is commonly described as "straight refinery" gasoline. This type of gasoline, as ordinarily refined, is colorless and possesses an odor varying from actual sweetness to the slightly disagreeable quality characteristic of kerosene. The hydrocarbons of straight refinery gasoline are of the so-called saturated class and are inactive when treated with such reagents as bromine, iodine and cold concentrated sulphuric acid. They are, however, slightly attacked by fuming sulphuric acid of moderate strength and by nitro-sulphuric acid mixture.

Straight refinery gasoline may vary within wide limits in the property of volatility but the distillation curves obtained by plotting the results by the Engler test have a more or less characteristic form. Fig. 1 shows the distillation curves of three gasolines of various degrees of volatility obtained from different types of crude petroleum. The percentage marks are plotted as ordinates and temperatures as abscissas. The curves slope upward rather sharply from the initial boiling point to approximately the 20 per cent mark. From this point the graph proceeds as practically a straight line sloping gradually upward to about the 90 per cent mark. Here another steeper slope commences, which generally proceeds to about the 95 per cent mark, and then breaks sharply. This break is due to the fact that the dry point is not actually the 100 per cent mark.

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Gasoline from Cracking Processes

The characteristic properties of cracked gasoline are odor and content of unsaturated or olefin hydrocarbons. Untreated cracked distillates have a particularly rank and unpleasant odor, which can however be practically eliminated by treatment with a small percentage of sul-

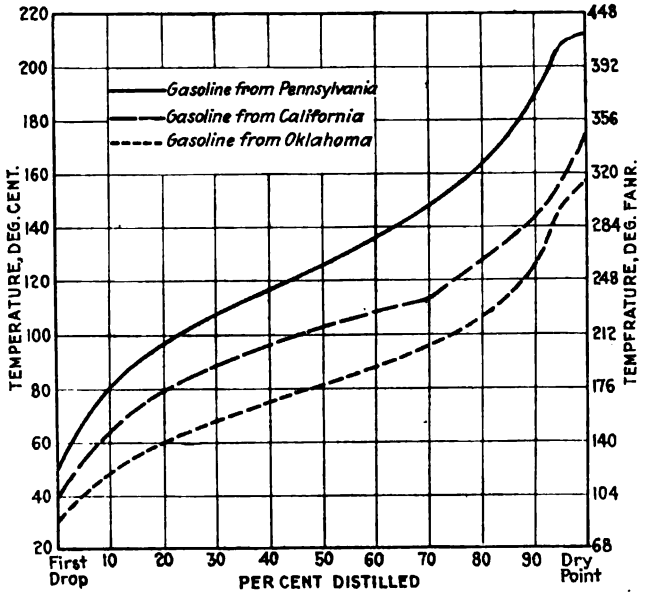

Fig. 1—Distillation Curves of Different Gasolines

phuric acid. The odor of a well-refined cracked gasoline is not noticeably disagreeable but has a definite quality and pungency that distinguishes it from gasoline of the straight refinery type. This odor is admitted to be somewhat undesirable but is not a serious objection. The other characteristic property of cracked gasoline is a considerable content of unsaturated hydrocarbons, compounds that react with such chemicals as bromine, iodine or concentrated sulphuric acid.

The amount of material removable from the gasoline by treatment with an excess of cold concentrated sulphuric acid is generally called the percentage of unsaturation. The cracked gasolines usually marketed are acid refined and blended with straight run and casinghead products and generally do not contain over 6 per cent of

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hydrocarbons removable by sulphuric acid. The volatility curves of cracked gasolines are, like those of straight-run products, controlled by the method of refining but the characteristic general form is the same as for the refinery distillates. Fig. 2 shows the distillation curve of an unblended cracked gasoline compared with that of a straight refinery product that happens to be of the same average volatility. The form of the two curves is almost identical.

Casinghead or Natural Gas Gasoline

Natural gas gasoline is of such a volatile nature that it is usually blended with a considerable proportion of heavier gasoline to reduce its vapor pressure so that it can be shipped and used safely. Its chemical and some of its physical properties are identical with those of the straight refinery type of product. It is colorless, of sweet odor, and inactive with sulphuric acid, bromine and iodine. The volatility of unblended casinghead gasoline is a property concerning which the Bureau has few data. This product is rarely used as such except perhaps in gasoline gas machines. It is usually marketed as a blend, either with a heavy naphtha produced by the ordinary refinery process or with straight refinery gasoline of high endpoint.

The term naphtha is used to designate a distillate which does not contain the low boiling fractions included with ordinary gasoline. The blending of casinghead gasoline with the straight run product is generally done in refineries, many of which make a practice of buying all of the former they can obtain and adding it to the distillates produced from crude petroleum. The blending of casinghead gasoline with naphtha is generally done either by the companies that produce the former from natural gas or else by blending companies.

Fig. 3 shows the characteristic distillation curves of blended casinghead gasoline as compared with the typical curve of a straight run product. One of the curves represents a sample of fuel that was obviously prepared by blending approximately equal quantities of casinghead and kerosene. This is a "shocking example" of what a blended gasoline ought not to be and is the only sample of this type of fuel that the Bureau of Mines has ever discovered for sale in the market.

Two of the other curves represent blends, one with heavy refinery gasoline, the other with naphtha. The naphtha blend has a lower initial boiling point and a higher content of volatile constituents than the gasoline

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blend. Both are fuels of good quality and are probably superior in starting properties to the straight run gasoline represented by the distillation curves shown for comparison.

Desirable Properties of Gasoline

The desirable properties of gasoline and some laboratory tests and specifications for determining them have been discussed in a recent technical paper* published by the Bureau of Mines. This paper was intended chiefly for the information of refineries and chemists but a brief review of some of the considerations mentioned may not be out of place here.

The requirements for a satisfactory fuel are as follows:

(1) That it shall permit ready starting of a cold engine;

(2) That it shall permit smooth and reliable operation of an engine at varied and rapidly changing loads and speeds;

(3) That it shall not contain substances such as free acid which by corrosion or otherwise injure an engine;

(4) That there shall be a minimum tendency for either solid or liquid residues to accumulate in the engine.

It is also desirable that gasoline shall not be of such nature as to be subject to excessive evaporation losses in storage or handling and that it shall not be too dangerous. These latter requirements are indefinite, however, and vary with climatic conditions. In addition they are likely to interfere with the starting qualities of gasoline if satisfied too thoroughly.

The second and third requirements are generally satisfied by present refinery practice.† The Bureau has heard no complaints either of corrosive action due to free acid in gasoline or of lack of flexibility in engine operation. The first requirement, easy starting of cold engines, seems to be least generally met, as the present tendency is to market gasoline of relatively low average volatility. There seems to be little possibility of improvement in this quality and the problem of easy starting will undoubtedly have to be solved by engine designers.

*Dean, E. W., Motor Gasoline, Properties, Laboratory Methods of Testing, and Practical Specifications. Bureau of Mines technical paper 166 (1917), 27 pages

†Since the publication of the paper evidence has been secured which indicates the occurrence of corrosion in engines, due to improper refining of gasoline. This phenomenon is now being investigated by the Bureau.

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The requirement that gasoline shall not leave a solid or liquid residue in the engine is one that frequently is not satisfied, according to reports from many makers and users of engines. Complaints are heard to the effect that there are sometimes gummy deposits in carbureters and manifolds, that excessive carbon deposits collect in cylinders, and that heavy fractions of gasoline are unburned and forced past cylinder rings into the crankcase oil.

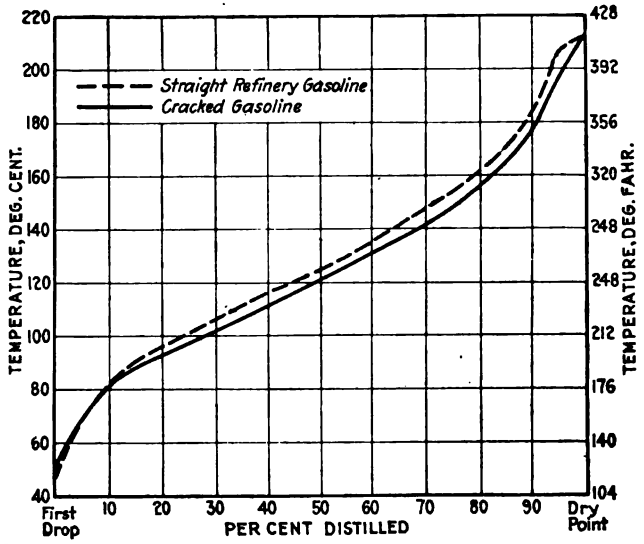

FIG. 2—DISTILLATION CURVES FOR CRACKED AND REFINED GASOLINES

Difficulties are without question frequently due to improper engine adjustment and operation, but the gasoline is undoubtedly to blame in some cases. The accumulation of gummy deposits in carbureters and manifolds is a type of trouble for which there does not at present seem to be any definitely established explanation. The Bureau is at present studying this problem but has not yet enough information at hand to make it desirable to attempt an adequate discussion. Fortunately this type of engine trouble does not seem to be of frequent occurrence.

Types of Cylinder Residue

The two most important types of residue are carbon, which collects in the explosion chamber, and heavy fractions of the gasoline that condense and escape past the

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piston rings into the crankcase. Carbon deposition is frequently due to improper carbureter adjustment and can generally be minimized if a sufficiently lean mixture is used. Such deposition is, however, in common with crankcase oil pollution, often due to the use of gasoline containing too great a proportion of heavy ends, which in everyday language means kerosene. The term used by the refiner and oil chemist is "high endpoint" and means that when the gasoline is subjected to a distillation test the presence of high-boiling fractions is indicated. These fractions undoubtedly are not vaporized even in cases when no fuel troubles are experienced but are burned while in the form of a fine spray, in the same way that coal dust or grain dust burns explosively.

Fuel trouble due to high endpoint occurs when heavy constituents are present in excessive proportion, so that instead of holding up in spray form drops of considerable size precipitate. These settle on cylinder walls and piston heads and cause trouble in a variety of ways. Lubrication is interfered with, there is deposition of carbon through cracking, and liquid residues may be forced past the piston rings and mixed with the lubricating oil. There may be other ways in which trouble is caused but those already mentioned are sufficient to indicate the disadvantage attendant upon the use of gasoline of too high endpoint in the average engine.

Unfortunately, it is a difficult matter to set any kind of definite limit that serves to indicate when the danger line in endpoint is passed. There are on record cases which seem to indicate that trouble has been caused by the use of gasoline of endpoint lower than 400 deg. fahr. On the other hand a tremendous quantity of gasoline is being used, generally without more than imaginary fuel trouble, which has an endpoint as high as 450 deg. In some cases 500 deg. endpoint gasolines are satisfactorily employed and the author is acquainted with one man who during the summer months regularly mixes with his gasoline a fourth of its volume of kerosene and obtains satisfactory results.

There is no question upon which it is easier to start an argument than on the proper endpoint of gasoline. The Bureau of Mines has up to date taken the stand that the only evidence it could depend on is the general marketing practice of refiners and on this basis it appears that the maximum satisfactory endpoint for a properly balanced gasoline is between the limits of 400 deg. and 450 deg. fahr.

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Objections to Blended and Cracked Products

There is still a tendency on the part of some users to consider that the only desirable variety of gasoline is a highly volatile product obtained by the straight refinery method. Blended casinghead gasolines are claimed to be dangerous, to deposit carbon and to stratify or change their composition in different parts of a storage system. Cracked gasolines are believed to be of vile odor and to polymerize in carbureters and manifolds and to leave ex-

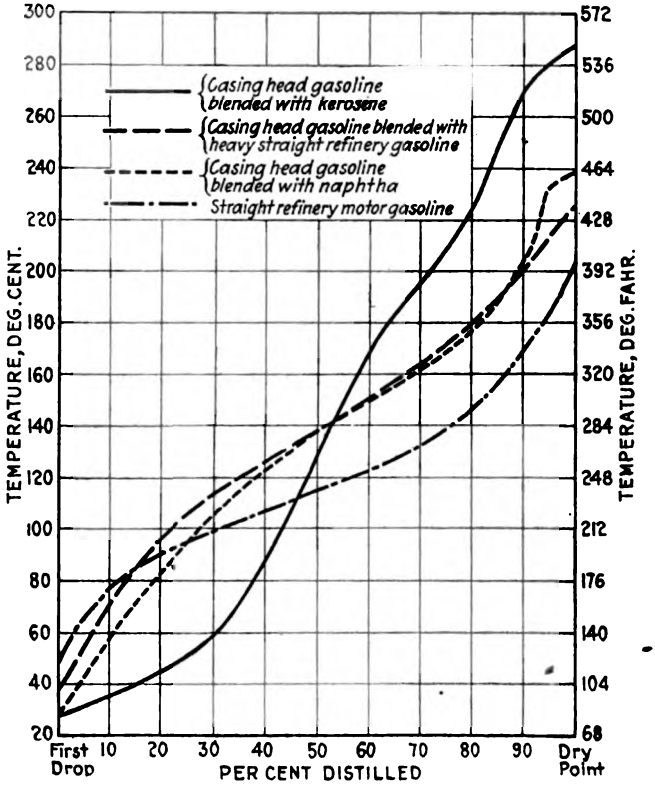

Fig. 3—Blended Casinghead Gasoline Compared with Straight Run Product

cessive carbon deposits in cylinders. Straight refinery gasolines of high endpoint are likewise condemned on the ground that they are bad in starting an engine and that they deposit carbon and pollute the crankcase oil.

These beliefs are not without some foundation of fact.

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There is little doubt that from the point of view of the average user the most satisfactory fuel ever produced was a straight refinery gasoline testing 70-deg. Baumé or higher. This type of product was relatively dangerous in handling and was subject to high evaporation losses but these disadvantages were ones to which users of the past were educated and which were considered characteristic of gasoline. The fact that gasoline was treated with proper respect minimized in a large degree the first disadvantage, and its relatively low cost minimized the second. Gasoline of this type worked satisfactorily under almost any sort of engine conditions and in any event was rarely blamed for trouble. In earlier days the custom when anything went wrong was to swear at the engine rather than at the fuel.

When the newer types of gasoline were first introduced the trouble and dissatisfaction that immediately arose was not without some just cause. In many cases the process of blending casinghead gasoline was not properly conducted and in any event 70-deg. Baumé gasolines of this type could not help being dangerously volatile. Methods of refining cracked gasoline were not perfected when these distillates were first marketed and they certainly did not compare favorably with straight run products in the matter of odor. It is likewise possible that there may actually have been cases of polymerization with troublesome deposition of gummy resinous material. When heavy straight refinery gasolines were first used engines were not designed or adjusted for their consumption and difficulties naturally arose. At present the methods employed in production and refining have been so improved that the bulk of the gasoline now on the market is really of excellent quality.

Both cracked and natural gas gasolines are put out in the form of blends that the average user cannot distinguish from straight refinery products and which have actual elements of superiority over the latter as now marketed. Heavy gasoline is, of course, deficient in the property of easy starting, but this objection is balanced by the fact that it actually contains more heat units per gallon and permits the development of more power. There are certain other admitted objections to relatively non-volatile gasolines, but these are chiefly of the sort that can be obviated by proper engine design and adjustment. The Bureau has yet to receive proof that a gasoline of 400-deg. fahr. endpoint and a properly balanced range of volatility (which means that from 15 to 20 per cent of constituents distill below the boiling point of water and

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that the other fractions are reasonably well distributed) has failed to yield satisfactory results in a well-adjusted automobile engine of up-to-date design. In many cases gasolines of as high endpoint as 450 deg. fahr. are being used with satisfaction.

The main advantage of the present type of fuel is that its introduction has permitted the producers to keep pace with the demands made upon them. If the older standards of quality had been maintained prices would have soared far above their present high level and in addition there would have been today an actual shortage. Few users of automobiles would prefer limitation in their supply of fuel to the slight inconvenience of having to treat their engines with the care and respect these complicated mechanisms actually deserve.

Future Developments of Fuel Situation

It is a difficult matter to make accurate predictions for the future of a situation involving as many variables as does the fuel market. The Bureau of Mines is able to obtain reliable information concerning gasoline production and quality but realizes its inability to foresee all factors governing market conditions. It is therefore necessary to be conservative in attempting to tell what is likely to happen in any limited region or at any given time.

The one definite prospect for the immediate future is that there will be a considerable withdrawal of fuel from the home market for use in military operations. The quantity needed can be conservatively estimated for the present calendar year at about a fifth of the total production of the country. There is a likelihood that the quality will also be changed in the direction of further decrease in volatility because gasoline for military purposes will necessarily be of higher test than the present average commercial grade. If the total production is to be kept up to the present mark this will mean that domestic gasoline will be of lower volatility than now. In addition, refiners may be obliged to curtail their use of sulphuric acid, which will result in the marketing of gasoline of inferior color and odor.

The fuel situation is unfortunately one for which the ultimate successful termination of the war does not promise permanent relief. Estimates as to the number of years our resources of crude oil will last have been variously made and it is agreed that unless unexpected resources exist another generation will witness an actual

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shortage in the raw material from which gasoline is now produced. It is certain that the world cannot go on forever using up crude petroleum at the present rate of consumption and it is even now desirable to face this prospect squarely and make allowances for what the future is likely to bring. The possibilities of new supplies of fuel will therefore be discussed briefly, with the idea of showing the relative importance of the sources of relief mentioned.

Use of Alcohol

When looking for a future supply of any product it is natural to consider raw material that can be continuously produced instead of taken from a stored up deposit. Alcohol is a volatile liquid fuel that is known to be satisfactory for use in internal combustion engines and which can be produced by the simple process of fermentation from any source of sugar. Unfortunately, however, evidence is still lacking that alcohol can be produced sufficiently cheap enough to compete with various other forms of fuel that come to us as it were ready-made. Just at present alcohol sells at a price that almost prohibits its use even as an anti-freeze compound. This situation is of course abnormal and there is every prospect that in the future alcohol may be on the market at a lower price than ever in the past.

The economical utilization of alcohol, however, requires engines that operate at compression pressures far greater than now employed for gasoline, which fact introduces complications that relegate alcohol to the far distant future as a potential substitute for gasoline. For the present it seems that alcohol is only of academic interest as fuel except perhaps in localities where gasoline is expensive and scarce and the most promising raw material for alcohol manufacture is cheap. Such a situation already promises to develop in the Hawaiian Islands; gasoline is obtained there only after shipment over sea, but low-grade unrefinable molasses is produced in large quantities.

Use of Coal-Tar Distillates

Another source of fuel that has received much attention is the by-product coking industry. The light oil constituents obtained by this process, such as benzol, toluol and xylol, are known to be satisfactory fuel, and although during the war the demand for them will continue to be greater than the supply, they will undoubtedly be marketed as fuel after the present abnormal pro-

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duction of explosives decreases. The use of these hydrocarbons either as such or blended with petroleum gasoline may increase the supply of fuel to a certain degree, but unfortunately the total available supply is relatively small. The following statistical information is interesting: The gasoline production of the United States for the calendar year 1916 has been variously given at from 2 to 2¼ billion gallons. The U. S. Bureau of Mines report on refinery operations for the year 1916 gives 2,058,-

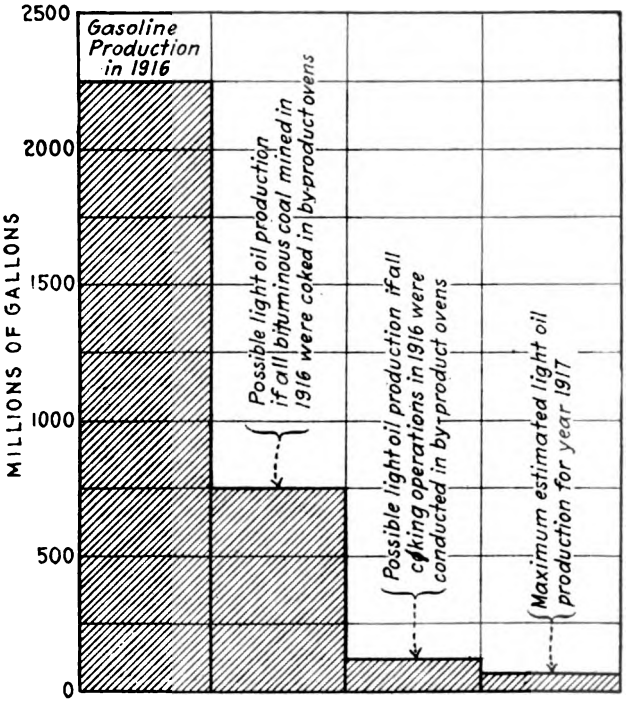
2500
2000
1500
1000
500
0
MILLIONS OF GALLONS
Gasoline Production in 1916
Possible light oil production if all bituminous coal mined in 1916 were coked in by-product ovens
Possible light oil production if all coking operations in 1916 were conducted in by-product ovens
Maximum estimated light oil production for year 1917

FIG. 4—FUEL FROM COAL COMPARED WITH ACTUAL GASOLINE PRODUCED

880,596 gal. This includes a portion of the production of casinghead gasoline, which has been given as 104,212,809 gal. by J. D. Northrup, U. S. Geological Survey press bulletin No. 332 (August, 1917).

The production of coal-tar light oils for the same year was 36,600,000 gal. (C. E. Lesher, *Iron Age*, vol. 99, Feb. 22, 1917), and it has been estimated that during the year 1917 this production will have been increased to from 40 to 60 million gallons. These figures indicate that if

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the total production of coal-tar light oils of the country were added to the engine fuel supply the increase would be about 2 per cent. These figures represent coking operations already conducted under the by-product method. Figures based upon the total coke production of the country and even the total coal production of the country are not much more promising. The total coke produced in 1916 by both beehive and by-product ovens is given as 54,533,585 short tons and the total bituminous coal as 502,519,682 tons. (United States Geological Survey press bulletin No. 343, November 1917.) It is likewise recorded that the average yield of coke from a ton of coal is 1440 lb. and the average yield of light oil is 1.54 gal. (C. E. Lesher, Mineral Resources of the United States, 1916, part 2, pp 515-558.)

Calculations from these figures indicate that if all the coke produced in the country in 1916 had been made by the by-product process, the 75,741,053 tons of coal used would have yielded 117,641,232 gal. of light oil, which if entirely devoted to use in internal-combustion engines would have increased the available supply of fuel by a little more than 5 per cent. If all the bituminous coal produced had been coked and had yielded the same average quantity of light oil as the grade of coal now used for coking the light oil produced would have amounted to 753,779,523 gal., which proves to be less than 34 per cent of the gasoline production. These statistics show the limitations of the coking industry as a source of engine fuel. Fig. 4 represents graphically the theoretical quantities of fuels produceable from coal as compared with the gasoline actually produced. The use of coal-tar light oils as fuel will undoubtedly aid to increase the supply at some time in the future, but they by no means promise to meet the total need for more gasoline.

New Resources of Crude Petroleum

It is by no means certain that the tremendous quantities of crude petroleum at present counted on exist, but producers are hopeful that such fields as the Mexican and even the Central American and South American fields may ultimately develop to a considerable degree of importance. The discovery of more crude petroleum would seem to be the easiest and most pleasant way of meeting the threatening fuel shortage, but this is a possibility that is so uncertain that it is hardly wise to place dependence on it.

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Gasoline from Oil Shale

Vast deposits of oil-bearing shale are known to exist in certain parts of this country and engine fuel can undoubtedly be obtained from this source. Here, however, as in the case of alcohol, the factor of expense in producing puts these products in the class of a resource that may be utilized in the future. Scientists are already giving attention to the problem of working up these shales and it is hoped that satisfactory methods for obtaining fuel in maximum quantity and of the best possible quality will be developed by the time it becomes commercially practicable to use the shales. Many problems in the fields both of distillation of the shales and treatment and refining of the oils remain yet to be solved, but these problems are considered to be comparatively simple of solution and are now being studied.

Wider Use of Cracking Processes

From the point of view of the oil refiner the most desirable way of increasing the supply is to get more of it from the same quantity of crude oil now refined. This can be accomplished either by extending the range of volatility acceptable for fuels or by the employment of methods that increase the quantity of volatile products obtainable from the crude. The latter possibility is that which the use of the cracking process covers. The development of these processes is today the most important field of experimentation in the industry and considerable progress has already been made. The Burton process, owned and controlled by the Standard Oil Company of Indiana, and operated by various of the companies of the Standard group, has been developed to a high degree of efficiency and is regularly converting large quantities of heavy petroleum distillates into satisfactory fuel.

The Hall process, which is said to be operated on a large scale in England and to some degree in the United States, is also apparently in the stage of successful commercial operation. The Rittman process, with which the Bureau of Mines has been largely concerned, seems now to be thoroughly developed for the large scale production of gasoline and should soon be a commercial factor in the market. Other processes now operating are the Greenstreet and the Cosden.

At present it appears that the Burton process and others of the pressure distillation type can operate successfully using as raw material fairly heavy petroleum distillates, such as the fuel and gas oil types. The Bur-

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ton process cracks a limited amount of kerosene also but it is not adapted in its present stage of mechanical development to the treatment of oils of this type. The Rittman and Hall processes are said to operate equally well with kerosene and the heavier distillates. The Greenstreet process seems to operate entirely with kerosene distillate. It is not known that any pressure cracking process is successfully utilizing oils of the residuum type in large-scale apparatus. These crack readily enough but the mechanical difficulties incident to the deposition of large quantities of carbon have so far interfered with commercial success.

Profitable Cracking Difficult

The general consensus of opinion of those qualified to judge is that the production of volatile hydrocarbons, satisfactory for use as fuel, can be brought about by many modifications of the principle of applying heat and pressure but that the cracking process in general cannot be made a commercial success unless handled with a high degree of scientific and technical skill and under carefully regulated economic conditions. In other words, although oil is easy to crack it is not easy to crack profitably. The oil cracking art is in many ways like the poultry business. Any amateur can convert grain and meat scraps into chickens and eggs but it takes more than interest and ambition to accompany this process by the accumulation of financial profit. Likewise, it is tolerably easy to make gasoline out of heavy oil but a totally different matter to keep the cost of the finished product satisfactorily below its market price.

It appears therefore that although the cracking process offers great possibilities of increase in the supply of fuel, this increase is bound to be a gradual development and in no way revolutionary. If cracking processes were used to the full extent of their applicability the total production of gasoline from the quantity of crude oil now produced might be doubled. This increase cannot, however, be expected on the basis of reasonable possibility in less than ten years.

An adequate discussion of cracking processes would require more time than automotive engineers would wish to devote even to this vital subject and for the present the situation can be summarized by stating that this method of increasing the fuel supply seems just at present to be considerably more important than any or all of the possibilities already discussed.

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Use of Heavier Petroleum Distillates

The possibilities of increasing our future supply of fuel have already been discussed from the points of views of several industrial divisions. It has been shown that the men in the producing end of the oil industry may help solve the problem by making available new supplies of crude petroleum. Scientists have the problem of making available the supply obtainable from oil shales.

The coal distillation and the fermentation industries may in the future supply greater or less quantities, and the refiners by improvement in their present methods and the development of cracking processes can contribute enormously to the country's production. There remains for discussion the part that designers and builders of internal-combustion engines have to play in increasing the potential supplies.

Increase in Engine Efficiency

It may seem presumptuous for a rank outsider in the gasoline engine business to suggest that one of the most important lines of activity for automotive engineers is to increase the potential efficiency of the engines they turn out. It actually seems, however, that there are tremendous possibilities in this line, not necessarily through increases in mechanical efficiency, but by educating the public to demand and use automobiles that are essentially economical in consumption of fuel. There are now on the market types of cars that on average roads and grades can travel about 20 miles on a gallon of gasoline. Most of these cars are of low and moderate price and it appears that the majority of higher priced cars get notably less mileage, this in some cases being as low as 6 or 8 miles per gallon.

Such a condition should not continue, because if fuel, at any price, is not going to be available, the automobile business is bound to suffer. For this reason it seems that designers and manufacturers ought to take this matter in their own hands, regardless of the present demands of buyers, and work out a reasonable standard of performance for all cars. For instance, cars might be built with the basic assumption that a two or three-passenger car must travel 22 miles on a gallon of gasoline, a five-passenger car 18 miles and a seven-passenger car 16 miles. Some such standard of efficiency as this should be regarded as ironclad and all automobile design based upon it. Such considerations as speed and hill-climbing power should not be allowed to interfere with this mat-

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ter of efficiency and the users will soon get over the idea that it is necessary to have 80-hp. engines in their cars when 30 hp. is sufficient to carry them at adequate speed anywhere they want to go.

Perhaps no phase of the fuel situation has so interested automotive engineers as the use of kerosene in place of gasoline. Present market conditions are such that kerosene is one of the cheaper petroleum products, and as it has already been demonstrated to work satisfactorily in internal-combustion engines under certain conditions, there has been a great desire to render it available for general use in automobiles. The Bureau of Mines has had called to its attention many devices for the utilization of kerosene but believes that mechanical development in this particular line is a mistake. The logical and reasonable way to utilize kerosene is not as such but as a mixture with the gasoline produced with it in the refining of crude oil. In other words, attention should not be given to the utilization of kerosene but to the utilization of petroleum distillates containing both the gasoline and kerosene fractions of crude oil.

In terms of the testing laboratory this means fuel of high endpoint and correspondingly lowered average volatility. Such fuels could be produced by the refiners with slight modification of their present methods and equipment, and their price should be somewhat less than the average price of gasoline and kerosene now marketed separately. The gain would be the simplification resulting from the production and handling of one instead of two products. The Bureau is not attempting to tell just how the problem of designing engines that can handle fuel of from 550 to 600 deg. fahr. endpoint can be solved, but accomplishment of such an end ought not to be beyond the skill and ingenuity of engine designers, especially as they have already seriously undertaken the infinitely more difficult task of using straight kerosene.

The utilizing in automobile engines of a type of fuel that contains the combined gasoline and kerosene fractions of crude petroleum would certainly double the available supply, and even if all the other agencies working to this end failed, the automotive engineers themselves could postpone for a long time the fateful day when their industry will be handicapped by not having material available to feed engines after these have been produced.

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Author's Conclusion

The present paper has developed along rather heterogeneous lines but can be summarized simply. The present fuel situation, in spite of the fact that gasoline is high in price and relatively low in volatility, is good. This will be appreciated more fully if war conditions add one more inconvenience to the already lengthy list that the country is experiencing. The newer types of gasolines are not inherently worse than the old; the price is not out of proportion to the price of crude petroleum, and the supply is abundant and well distributed.

In the future it appears that oil producers must secure more crude petroleum and refiners must get greater yields of volatile products from the oil treated. Designers must work toward higher standards of fuel economy and must strive to develop engines that will handle fuels of lower average volatility than the present type. A certain increase in supply of fuel may be counted on from the coal distillation industry and eventually the supply may be derived either from oil shale or from fermentation industries.

In closing it is desired to remind the members of the Society of Automotive Engineers that the present paper makes no pretense to being a last word on the important subject with which it deals. The various phases of the fuel situation could be discussed at almost infinite length and it is realized that even the few problems mentioned in this paper have not been given the detailed consideration they deserve. It is hoped, however, that the attention of the engineers of the Society may have been called to some problems that they can profitably study.

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HEAT-FLOW THROUGH CYLINDER WALLS

By Louis Illmer*

Modern requirements have already forced the rotative speed of high-duty gas and oil engines to a point where the difficulty of heat-flow control, especially with cast iron cylinders, tends to arrest further progress in this direction.

In view of this inherent limitation the art of high-speed engine design can best be advanced, not by continued experimental exploration, but rather by first establishing the basic principles underlying heat-flow effects.

The purpose of the present paper is to demonstrate that every internal-combustion engine of given size and type has a safe speed limit and that this can be predetermined upon a rational heat-flow basis. This paper provides an explicit method of procedure, by means of which the design characteristics of a normal gas or oil engine can be critically analyzed for heat-flow effects.

In addition, the matter of relative heat-flow in two versus four-stroke cycle engines, which has been the subject of much controversy, is investigated and certain conclusions are drawn as to the merits of each type.

Introduction

The fixing of a suitable margin of safety for a steam-engine cylinder is a comparatively simple matter when the working pressure and strength of material are known. In the case of gas-engine cylinders, on the other hand, the stress relations become more involved because of the additional temperature stresses set up by heat-flow through the cylinder walls. The faster the engine speed, the greater will be the temperature stress produced.

This paper undertakes to analyze the combined pressure and temperature stresses in gas-engine cylinder walls and to prescribe the limiting stresses and extreme engine speeds that will insure satisfactory operation. The method pursued is that of fixing the maximum allowable rate of heat-flow into the jacketed walls of any given cylinder from jacket-loss determinations.

The two principal factors that impose restriction upon rotative speeds are excess heat-flow and undue inertia pressures. The thermal aspects of such limitations will

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be considered only, since the inertia effects can usually be kept within bounds by suitable construction.

A considerable number of the available jacket-loss tests have been analyzed for heat-flow data, but few if any investigators report all their data in the detailed form desired for present purposes. None of such tests was conducted throughout a sufficiently wide range of conditions to define fully the controlling factors of heat-flow. It was necessary therefore to pick out portions of data furnished by different experimenters and to work for a composite result. The conclusions thus drawn are thought to be sufficiently accurate to fix the speed limitations imposed by excess heat-flow, and to indicate means by which its destructive effect can be minimized.

Furthermore, this analysis of jacket-loss tests has been supplemented by a study into the speed limits of engines as used for various kinds of commercial service. By collecting a sufficiently large amount of such data, a fairly satisfactory value was determined for the limiting rotative and piston speed for each particular kind of engine. Finally, by combining these separate constants on the basis of the heat-flow laws deduced from jacket-loss tests, it was possible to arrive at a basic formula for speed limits, applicable to small high-speed engines as well as to large slow-speed engines of maximum output.

Synopsis of Heat-Flow Formulas

For the sake of a clearer presentation, certain conclusions will be briefly reviewed before citing the data upon which the given heat-flow rate has been based.

As found by the methods described, three principal factors control heat-flow through the jacketed cylinder-walls of internal-combustion engines, and these can be expressed thus:

$$H = C_f \left(\frac{MEP}{E}\right)^{1/3} \frac{N}{\log_{10} N} F^{2/3} \qquad (1)$$

where H = uniformly distributed full-load heat-flow through the jacketed cylinder-wall as measured in B.t.u. per hour per sq. in. effective cooling surface A_e.

C_f = constant depending upon the cycle of the engine.

MEP = mean effective pressure, lb. per sq. in.

E = thermal efficiency as indicated in the power cylinder.

N = engine speed, r. p. m.

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$log_{10}N$ = common logarithm of N.

F = surface factor, that is, effective volume V_e in cu. in., divided by the effective surface A_e in sq. in., as measured with the piston standing at one-fourth stroke from its inner dead-center position.

As given, (1) is intended to apply only to engines operating at or near rated capacity. It will be seen that heat-flow is largely dependent upon the factor MEP/E, which in turn is determined by the heat-input per cubic foot of piston displacement, thus:

$$\text{Heat input/cu. ft. displ.} = \frac{144\,MEP}{778\,E} = 0.185\,\frac{MEP}{E} \quad (2)$$

The heat-input measures the temperature rise occurring in the working charge during the explosion period. Furthermore, the ratio MEP/E can, without serious error, be used to approximate the average temperature of expansion; for explosive engines its value usually lies between 8 and 10, and averages about $9MEP/E$.

The first factor of (1) can be considered as the temperature factor of heat-flow. The value of the exponent k of the factor $(MEP/E)^k$ is found to lie between 5/4 and 3/2, with the given value of 4/3 as a fair average for full-load conditions.

Thus, the rate at which heat flows from the hot working charge into the surrounding cylinder wall is found to be dependent in part upon the average temperature head maintained during the expansion stroke as measured with respect to the absorbing jacketed wall, and partly, in my opinion, upon the greater thermal conductivity of the working charge at high temperature.*

The absorption of heat during the suction and compression strokes of a four-stroke cycle engine can be taken as negligible. The comparatively small jacket loss occurring during the exhaust stroke is most conveniently taken in terms of the heat loss of the expansion period.

The second factor $N/\log_{10}N$ of (1) is a time factor. The time of action for a single explosion is measured in

*According to O. E. Meyer's "Kinetische Theorie der Gase," the thermal conductivity of a gas at high temperature can be expected to increase in the same ratio as does the product of coefficient of internal friction (viscosity) and its specific heat. Assuming the latter to remain constant, the thermal conductivity of air, for instance, as based upon Barus' viscosity experiments, is found to be about 4/3 larger at 3000 than at 2000 deg. fahr.; hence, within the usual range of expansion temperatures, the conductivity of air changes approximately as the two-thirds power of the temperature increase. While the corresponding conductivity factor, as fixed for internal-combustion engines by (1), increases at a somewhat slower rate, this difference is no doubt chargeable to the altered character of the working charge and the approximate method used for estimating temperatures.

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terms of $1/N$ minutes, but its relative effect in driving heat into the cylinder jacket is found to be proportional to $1/\log N$.

The speed of 10 r.p.m. has been selected as a convenient unit of time reference. Thus, at 1000 r.p.m. the relative heat-absorbing capacity per cycle of a given cooling surface is about one-third that at 10 r.p.m. The cyclic heat-absorbing capacity multiplied by the number of temperature applications N provides a measure for the total heat-flow into the cylinder wall, in so far as speed effects are concerned.

The last factor F of (1) is the surface factor. When a small surface surrounds a large volume, the rate at which heat will be given off to a unit surface must necessarily be more intense than from a small volume enclosed by a relatively large surface.

The variation in jacket loss that occurs with change in the surface condition can be taken care of by the surface factor F, as indicated in (1). The numerical value of F is dependent upon the shape and dimensions of the combustion chamber, and for similarly built engines it increases directly with the cylinder-bore dimension.

In determining the cubic content per unit of surface, not only must the enclosing head and piston surface be taken into account, but, to arrive at a correct value for the total effective cooling surface of the cylinder parts, a certain portion of the bore surface should also be included. Since all parts of the bore surface do not absorb heat from the working charge at the same rate per unit of exposed surface, the total cooling effect of the bore wall can conveniently be expressed in terms of an equivalent surface having the same rate of heat flow as the initial clearance surface.

Further experimental evidence is required to determine such a bore allowance accurately, but from an analysis of the available data, it appears that about one-fourth of the cylinder-bore surface can be taken as a fair allowance for this added cooling effect. Accordingly, the surface factor F is to be measured by dividing the clearance volume plus one-fourth the piston displacement by the sum of the initial clearance surface and one-fourth of the bore surface, that is, V_c/A_c.

Order of Treatment

The further discussion of this subject will be treated under the following headings:

(A) Jacket-loss test data from which (1) was deduced.

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(B) Temperature stresses set up in cylinder walls by heat-flow.

(C) Pressure stresses produced within cylinder walls by the explosive pressure.

(D) Bore temperature formulas as based upon the temperature head required to drive a stipulated heat-flow through the cylinder wall and into the surrounding jacket-water.

(E) Rotative-speed limits as determined from engines in commercial service.

(F) Bore-temperature limits as deduced from engines in commercial operation.

(G) Formulas for the minimum cylinder thicknesses applicable to high-speed engines.

(H) Allowable heat-flow constants as based upon the prescribed bore-temperature limits.

(I) Formulas for the surface factor *F*.

(J) Limiting speed formulas for low-compression gas and oil engines as deduced for the prescribed bore-temperature limits.

(K) Two versus four-stroke cycle engines as viewed from the standpoint of heat-flow effects.

(L) Formulas for jacket-loss as measured in terms of heat input.

(M) Constancy of indicated efficiency with increasing jacket-loss.

(N) Importance of small surface factors and the advantages accruing from adequate cooling surfaces, especially for large engines.

(O) Recapitulation or conclusion.

Analysis of Jacket Loss Test Data

The jacket-loss tests made in an experimental research study by Gibson and Walker† provide the most suitable data with which to determine the first two factors of (1). These tests apply to a National 11 by 19-in. single-acting four-stroke cycle gas engine, arranged to alter the compression ratio between the limits 5.17, 5.70 and 6.62. The engine speed was also varied from 150 to 250 r.p.m.

The constructive features of this engine proved to be well adapted for consistent jacket-loss determinations. The cylinder head is cast integral with the cylinder proper, but the jacket space is divided into two compartments, one of which surrounds the exhaust passage. By this means it was possible to separate the exhaust loss

†Inst. of Mechan. Engrs., May, 1915. Distribution of Heat in the Cylinder of a Gas-Engine

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and to compute the jacket loss during the expansion stroke up to the point of release.

As the cited reference does not record all the desired data in a form best suited to present purposes, it has been necessary to interpolate a portion of the data, giving the results shown in Table I. The heat input has been estimated on the basis of (2), and when multiplied by the given jacket-loss percentage measures the gross heat-flow through the cylinder walls.

Although three sets of tests were made at the specified expansion ratios, the range of these experiments proved too close to bring out definitely the effect upon heat-flow of change in the expansion ratio R. Hence Table I analyzes only one of these tests, namely, that for $R = 5.7$. The agreement between the given empirical relations and the experimental data are thought to be sufficiently close to establish the first two factors of (1).

Surface Factors

Owing to the limited test data, it has been more difficult to arrive at an entirely satisfactory mode of treatment for surface effect.

Granting the correctness of the first two factors of (1),

TABLE I—ANALYSIS OF JACKET-LOSS TEST DATA

B.HP., Full Load	N R.p.m.	Air to Gas Ratio	MEP Lb./Sq.In.	Ind. Eff. E (%)	$\frac{MEP}{E}$	% Jacket Loss, J*	Heat-Flow, H*	$H \div \left(\frac{MEP}{E}\right)^{\frac{1}{3}}$		Constant C*
25.0	150	1:7	82.8	32.8	252	27.0	148	0.0932		
20.0	150	1:9	69.5	34.1	204	25.3	112	0.0933	0.0925	0.000822
16.5	150	1:11	61.1	34.1	179	23.5	92	0.0911		
31.5	200	1:7	81.9	33.5	244	25.3	179	0.117		
25.0	200	1:9	67.8	34.9	194	23.5	132	0.117	0.116	0.000818
20.5	200	1:11	58.6	34.9	168	21.5	105	0.113		
36.0	250	1:7	80.5	33.8	238	24.5	212	0.143		
28.5	250	1:9	67.3	35.4	190	22.8	157	0.144	0.141	0.000828
23.5	250	1:11	57.8	35.4	163	20.7	122	0.137		

*Jacket loss in terms of heat-input as measured to the end of expansion stroke equals about 80 per cent of total jacket loss.
Effective cooling surface Ae = about 400 sq. in.
Surface factor F = about 2.09.

an allowance for surface effect can be determined by analyzing a series of jacket-loss tests as made upon similar engines of widely different dimensions. The most suitable data that the author has been able to find for this purpose apply to some old-style, low-compression four-stroke cycle Deutz gas-engines.*

*Brauer, Z. d. V. D. Ing., 1887

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The results, set forth in Table II, show that the heat-flow per unit of effective cooling surface does increase to a marked degree with increased cylinder dimensions. On the basis of this and other data, the exponent n of the factor F^n has been placed at 2/3, as given in (1) and plotted in Fig. 1.

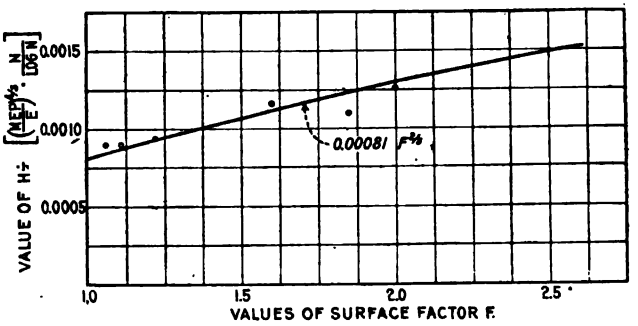

FIG. 1—CURVE SHOWING SURFACE EFFECT

Equivalent Surface Allowance for Cylinder Bore

The equivalent surface allowance that should be added to the clearance surface in order to allow for the cooling effect contributed by the cylinder bore can be found by working with an engine having provision for an extended change in the ratio of expansion. The most serviceable test for this purpose appears to be that made by Prof. F. W. Burstall.† This was made upon a 16 by 24-in. single-acting horizontal four-stroke cycle Premier gas-engine running at 170 r.p.m., with provision to change the ratio of expansions from 4.27 to 8.07. Furthermore, the exposed clearance surfaces are fully specified.

This engine was provided with a separate flat cylinder head bolted to a massive internal-jacket flange, thus forming the combustion chamber in an extended portion of the cylinder bore. The jacket losses appertaining to the head and the cylinder jacket, as well as to the water-cooled piston and valves, are reported separately.

In this test the head jacket shows a disproportionate percentage of jacket loss when measured in relation to its cooling surface, but this is probably due to the massive cylinder flange and the considerable conductive

†Third Gas-Engine Research Committee Report, Proc. Inst. of Mech. Engrs., Jan., 1908

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power of the cylinder-head studs. To use this test to advantage, it has been deemed advisable to work with average values of the given total jacket losses and to group the tests as indicated in Table III.

TABLE II—ANALYSIS FOR SURFACE EFFECT

Rated B.Hp. per Cyl.	3	3	4	8	12	16	25
Bore, in.	6.1	6.75	6.75	9.1	10.6	11.4	13.0
Stroke, in.	12.2	10.2	13.4	15.7	17.7	20.4	26.0
Rev. per min., N	182	178	160	159	140	141	139
Jacket loss, J, per cent	52.0	54.0	49.5	47.6	39.4	40.8	43.0
$M.E.P.$, lb./sq. in.	42.3	42.2	45.0	44.5	44.6	49.4	50.0
Ind. eff., E, per cent	18.2	18.4	20.3	19.9	20.7	22.4	23.7
MEP/E	232	229	222	228	215	220	211
Surface factor F, about	1.11	1.06	1.22	1.60	1.85	2.0	2.37
Heat-flow H, about.	94	94.5	84	107	85	101	118
$H+\left[\left(\frac{MEP}{E}\right)^{\frac{1}{2}}\frac{N}{Log\,N}\right]$	0.00090	0.00090	0.00094	0.00116	0.00110	0.00126	0.00157
Constant Cf	0.00084	0.00088	0.00083	0.00085	0.00073	0.00079	0.00090

The equivalent bore surface allowance can be found on the assumption that the heat-flow constant C_f applying to the respective cooling surfaces A_s must be about the same for each one of the different expansion ratios used. As shown in Table III, this condition is best satisfied when the equivalent bore-surface allowance is taken as constant, namely, at $\pi DL/4$, the bore and stroke being D and L, respectively.

An estimate of this bore-surface allowance can also be made by using equal crank angles as a measure of time, and then integrating the product of the respective time, temperature and surface factors corresponding to each such crank position. The resulting value for the equivalent bore allowance is somewhat larger than that determined upon the experimental basis.

While the available experimental data are not sufficiently complete to furnish conclusive evidence as to the exact laws for the surface influence, the described procedure does allow of drawing some important conclusions as to the manner in which heat-flow can be expected to affect speed limitation. It will now be shown that the formulas deduced give reasonably consistent results when applied to engine jacket-loss tests covering a wide range of conditions.

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TABLE III—CHECK FOR BORE SURFACE ALLOWANCE

Test No.	N R.p.m.	Compression		$M.E.P.$ Lb./Sq. In.	Ind. Eff. E (%)	$\frac{MEP}{E}$	Jacket Loss		Surface As, Sq. In.	Surface Factor F	Heat-Flow H	Constant Cf
		Lb.-Gage	Ratio R				Per Cent Input J	B.t.u. per Min.				
A & B	168	207	7.86	84	39.6	212	19.7	1802	896	2.13	121	0.00078
C	170	173	7.22	92	41.5	222	20.6	2012	913	2.18	132	0.00078
D	171	157	6.79	98	40.4	243	19.5	2108	927	2.19	138	0.00070
F	169	132	6.20	101	35.9	281	20.2	2633	952	2.24	166	0.00070
J	167	127	5.45	94	34.5	272	28.6	3311	971	2.35	205	0.00088
Q	168	88	4.36	87	30.0	290	27.4	3453	1077	2.45	193	0.00074

*Brauer. Z. d. V. D. Ing., 1887

†Third Gas-Engine Research Committee Report, Proc. Inst. of Mech. Engrs., Jan. 1908

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Engine Jacket-Loss Analyses

A set of variable speed tests showing the application of the principles deduced was conducted by Prof. W. T. Fishleigh and W. E. Lay* upon a 4⅛ by 5¼-in. four-stroke cycle six-cylinder L-head automobile engine. The check results are presented in Table IV.

TABLE IV—AUTOMOBILE ENGINE JACKET TEST—F=ABOUT 0.6

N R.p.m.	Approx. MEP, Lb./ Sq. In.	Ind. Eff. (E), per Cent	MEP/E	Per Cent Jacket Loss		Heat Flow H	Constant C_f	
				To Water	To Fan			
640	41	19.3	212	40	20	260	0.0009	
640	67	22.1	303	40	19	390	0.0012	
1000	42	18.5	227	31	18	372	0.0008	0.00095
1000	68	19.6	346	28	15	502	0.0009	
1350	44	19.6	224	29	25	557	0.0010	
1350	70	19.9	351	24	18	687	0.0009	

Applying these check principles to large gas engines, Gueldner† cites a number of tests that confirm the general deductions embodied in (1). Tests taken from this source are analyzed in Table V.

TABLE V—LARGE GAS-ENGINE JACKET-TESTS—S. A. FOUR-STROKE

	Westinghouse		Deutz 15x22⅞	Crossley 11½x21
	13x14	25x30		
R.p.m., N	258	150	205	180
MEP, lb./sq. in	83	75	67.2	102
Ind. eff. E, per cent	25.5	27.4	29.5	34.5
MEP/E	325	274	240	296
Jacket loss, J, per cent	32.1	33.3	35.5	.....
Heat-flow, H	425	255	211	190
Surface factor, F	2.4	2.9	2.0	1.7
Constant CF	0.00099	0.00102	0.0012	0.00085

A further test, as made by Professor Hopkinson‡ upon a 11½ by 21-in. single-acting four-stroke cycle Crossley gas engine, running at 180 r.p.m., is also analyzed in Table V. This test is of special interest because the heat-flow into the non-cooled piston head was experimentally determined by means of its temperature gradient.

*Heat Balance Tests of Automobile Engines. Part I, Vol. 10, S. A. E. Transactions

†Internal Combustion Engines. Trans. by Prof. H. Diederichs

‡Proc. Inst. of Civil Engrs., February, 1909, on Heat Flow and Temperature Distribution in the Gas-Engine

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The estimated heat-flow H into the cylinder walls of this engine, as given in Table V, is found to be about 4/3 larger than the determined rate of heat-flow into the piston head. About one-half of such retardation in heat-flow can be ascribed to the higher counter-temperature of the piston face, and the other half is probably due to the relatively weaker explosive mixture lying adjacent to the piston face. This restricted heat-flow has been allowed for in working out the accompanying tables by taking the equivalent effective piston area for single-acting non-cooled pistons at three-fourth actual area.

For the sake of comparison, similar checks have been made for heat-flow through the cylinder walls of single-acting four-stroke cycle Diesel oil engines. The data shown in Table VI have been taken from various sources.** It appears that, notwithstanding the con-

TABLE VI—ANALYSIS OF DIESEL ENGINE JACKET-LOSS TESTS

Make of Engine	Amer. Diesel	Augsburg	Augsburg	Carel's	British Diesel	Amer. Diesel
Size, Inches	10¼x16½	9⅛x15⅝	6½x10⅝	22x29½	26x29½	16x24
N, r.p.m.	192	154	256	153	150	167
MEP, lb./sq. in.	87.6	105	103	99	95	81
Ind. eff. E, per cent.	33.9	34.2	40.6	41	39.3	30.4
MEP/E	259	307	254	242	242	266
Jacket loss, J, per cent	.....	.....	21	21.5	22.1	26.2
Heat-flow, H	205	221	194	215	211	214
Surface factor, F	1.4	1.3	0.9	2.6	2.9	1.8
Constant C_f	0.0012	0.00126	0.0012	0.0011	0.0010	0.0011

trolled combustion, the heat-flow through Diesel engine cylinder-walls can be treated in substantially the same manner as for explosive engines.

Summary of Jacket Test Constants

A considerable variation in the value of the constant C_f is to be expected from experimental data taken haphazard and without sufficient detailed information to allow of an accurate evaluation of the surface factor F. The given formulas are sufficiently comprehensive, however, to cover a wide range of engine sizes, as is shown by the following summary:

**Catalogue of the American Diesel Engine Co., Inst. Naval Architects, 1914. Chalkley "Diesel Engines for Land and Marine Purposes." Jour. Am. Soc. Naval Engrs., November, 1905

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Table I Average C_f = 0.00103 for F = 2.09
Table IV = 0.00095 = 0.6
Table V = 0.00102 = 1.7 to 2.9

The average constant C_f as given above is derived from the total jacket loss. Of this about 10 per cent is a fair allowance for the relative heat-loss occurring during the exhaust stroke of a four-stroke cycle engine, while the jacket around the exhaust passage can be expected to absorb an additional 8 to 12 per cent of the gross jacket loss.

The exhaust-passage jacket-heat does not pass through the cylinder or head walls, but in order to allow for contingencies, the value of the effective heat-flow H for a four-stroke cycle engine will be fixed at:

$$H = 0.001\left(\frac{MEP}{E}\right)^{1/3} \frac{N}{\log_{10} N} F^{2/3} \qquad (3)$$

The corresponding value of the constant C_f that applies to two-stroke cycle engines is readily deduced from the following considerations:

Doubling the number of explosions affects only the log factor of (1), and this can be allowed for by increasing the constant to 16/9 C_f. Deducting 10 per cent from the four-stroke constant as an approximate allowance for the heat loss occurring during the exhaust stroke, the corresponding constant for two-stroke cycle engines becomes equal to 0.0009 $\times$ 16/9, thus:

$$H = 0.0016\left(\frac{MEP}{E}\right)^{1/3} \frac{N}{\log_{10} N} F^{2/3} \qquad (4)$$

Accordingly, the heat loss of a four-stroke cycle engine is about 5/8 that of a two-stroke cycle engine for identical speed, temperature and surface factors.

Comparatively few two-stroke jacket-loss tests are available with which to check the newly-found value of C_f as given in (4). Some commercial tests made upon a 20 by 36-in. double-acting two-stroke cycle engine of the author's design work out to C_f = 0.0015, as based upon the total head-end and crank-end jacket losses. Tests made upon a large Oechelhauser gas engine, as reported by Prof. E. Meyer, show a check value slightly below that given in (4). Hence the ascribed value for C_f should amply cover the heat-flow for two-stroke cycle engines.

Owing to the perceptible loss of stroke resulting from piston-controlled exhaust ports, the equivalent bore allowance for two-stroke cycle engines should be determined

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on the basis of the effective stroke when working out numerical values for the surface factor F. For single-acting engines of this type, the effective stroke becomes equal to about $\frac{7}{8}$ L.

Temperature Stresses

Having fixed upon suitable values for the heat-flow through the cylinder wall, an estimate can now be made of the resulting temperature stress. The temperature gradient set up by the heat-flow H passing through a cast iron cylinder wall can be taken as:

$$t_g = \frac{H}{K} = \frac{H}{3.1} \qquad (5)$$

where t_g = temperature gradient, or drop in deg. fahr., per inch of metal thickness.

K = specific thermal conductivity of the cylinder metal per inch of thickness in B.t.u. per hr. per sq. in. of section at 1 deg. fahr. head,

= about 3.1 for cast iron at normal bore temperature.

Accordingly, the temperature drop due to heat-flow through a cylinder-wall of thickness S is equal to

$$t_d = t_g S \qquad (6)$$

where S = uniform thickness of head or bore wall in inches,

t_d = total temperature drop in deg. fahr. through the cylinder wall.

Hence, when the wall of an engine cylinder is subjected to a heat-flow equal to H, the temperature assumed by the interior surface will be t_d degrees higher than the exterior surface. This temperature difference produces a compression stress at the interior surface of the bore wall approximately equal to the tension stress set up at the exterior surface, thus

$$f_t = \frac{t_d}{2} eM = \frac{t_d}{2} \times 0.000006 \times 13{,}000{,}000 \qquad (7)$$

$$= \text{about } 40 t_d.$$

where f_t = temperature stress in lb. per sq. in., set up by the heat flow H.

e = coefficient of linear expansion for hard cast iron.

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M = modulus of elasticity for hard cast iron at stress of about 8000 to 9000 lb. per sq. in.

Pressure Stresses

The cylinder wall is also subjected to a pressure stress, which, when added to the temperature stress f_t, gives the total stress. The combined tension stress must not be allowed to exceed certain well defined limits; otherwise fatigue of material will result.

The tension stress resulting from the explosive pressure is not distributed quite uniformly over the entire wall thickness, but as shown by Clavarino's formula,* the stress at the outer surface is somewhat lower. As based upon this formula, the tension stress at the exterior surface of a cylinder under internal explosive pressure can be taken as

$$= \frac{5}{12} \frac{P}{\frac{S}{D} + \left(\frac{S}{D}\right)^2}$$

Multiplying this by a shock factor of 4/3, the desired tension stress due to the explosive pressure can be found from

$$f_p = \frac{0.56P}{\frac{S}{D} + \left(\frac{S}{D}\right)^2} \qquad (8)$$

where f_p = pressure stress, lb. per sq. in., at the exterior cylinder surface.
P = explosive pressure, lb. per sq. in.
S = uniform thickness of the cylinder wall, inches.
D = cylinder bore, inches.

The total tension stress produced by the combined action of the temperature and pressure effects is measured by the sum $f_t + f_p$.

According to Stromeyer's tests,† a fully reversing load producing a stress of $\pm$ 9000 lb. per sq. in. can be indefinitely repeated upon a good grade of cast iron without producing any serious fatigue effects. For the limited cylinder stress range of $+f_t$ to $+(f_t + f_p)$, a good sound cast iron cylinder should therefore withstand a stress of, say, 13,500 lb. per sq. in. Allowing for a minimum net factor of safety of at least 3/2 as measured with respect to the fatigue limit, this would reduce

*See Merriman: Mechanics of Materials
†Law of Fatigue—Trans. Inst. Naval Architects, Vol. LVII, 1905

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the maximum allowable working stress to about 9000 lb. per square inch. Whenever possible, and especially for engines that have to pull their rated load continuously for long periods of time, it is advisable to use a net factor of safety of 2, thus reducing the total safe stress limit for $f_t + f_p$ to about 7000 lb. per square inch.

Bore Temperatures

Knowing the temperature drop t_d as fixed by (6), it is now possible to estimate the maximum temperature assumed by the cylinder bore. The rate of heat-flow through the bore surface lying adjacent to the cylinder head is practically identical with that of the head wall itself.

Furthermore, the rate of heat-flow that applies to a gas engine cylinder is far in excess of the usual rate of absorption in steam boilers, which latter rate may be set at from 20 to 40 B.t.u. per square inch per hour. The far greater rate of heat-flow through the cylinder-head wall of a fast running engine, causes the jacket water immediately in contact with its exterior surface to reach a temperature at least equal to that at the boiling point.

In addition, a certain temperature head is required to drive the heat from the head surface into the boiling water. According to Marks,‡ such water takes up about 7 B.t.u. per square inch per hour for every 1 deg. fahr. temperature head as measured with respect to the boiling point. Making due allowance for the various factors noted, the bore surface and interior face of the

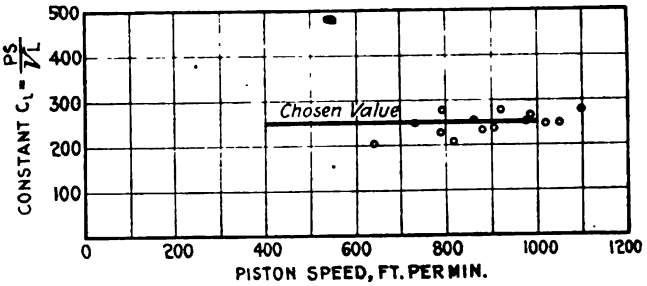

FIG. 2—VALUE OF CONSTANT C_1 FOR 2-STROKE CYCLE DIESEL ENGINE

cylinder head wall can be expected to reach a temperature equal to about

$$t_b = 212 + H/7 + (H/3.1 \times S) \qquad (9)$$

‡Mechanical Engineers' Handbook

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where t_b = probable maximum bore-temperature, deg. fahr., as reached in cast iron cylinders.

This equation applies only when the bore metal is uniformly disposed and is effectively water-jacketed throughout. The use of heavy internal cylinder-jacket flanges or any other excessive massing of metal is likely to cause additional localized heating of the cylinder bore.

Rotative Speed Limits from Commercial Engines

Having fixed upon the determining factors controlling heat flow and its effects upon the bore temperature, suitable working limits for these factors can now be found from a study of high-speed commercial engines. The limiting piston speed for any given kind of engine, such as submarine, cargo ship, motorboat, airplane, automobile, stationary and like engines, appears to increase more or less regularly with the stroke dimension. This relation can be approximately expressed by the following simple equation:

$$PS = C_l\sqrt{L} \text{ or } N = \frac{6C_l}{\sqrt{L}} \qquad (10)$$

where $PS = NL/6$ = piston speed, ft. per min.
L = piston stroke, inches.
C_l = piston speed constant dependent upon the type and intended service of the engine.

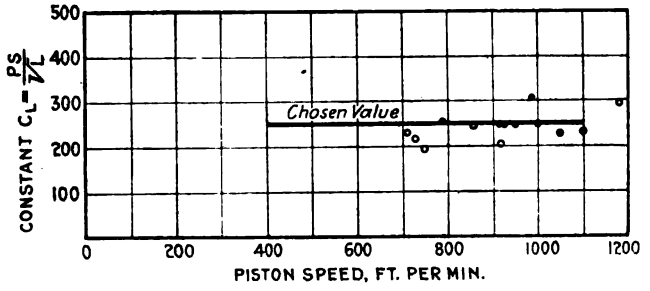

Fig. 8—Value of Constant C_l for 4-Stroke Cycle Diesel Engine

For any specified service, the allowable piston speed-limit is usually taken as proportional to the square root of the stroke. This investigation has further shown that each of the various kinds of engines has a critical piston speed limit that should not be exceeded for satisfactory operating results.

The values of the constant C_l as found for two and

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four-stroke cycle submarine Diesel engines are plotted in Figs. 2 and 3 respectively. Other kinds of engines were treated in a similar manner and the values chosen for the representative constant C_l applying to the various engine groups are given in Table VII. The results show reasonably consistent relative values when making due allowance for the limit of cylinder dimensions and intended service of each. The usual range of stroke-bore ratio is also given in the table.

Some interesting results can now be arrived at by analyzing the data in Table VII for heat-flow effects. As will be shown presently, the larger sizes of high-speed engines suffer considerably more from heat-flow effects than do the small fast running engines. The stroke dimension at which heat-flow can be expected to become a deciding factor in speed limitation for any one of the specified kinds of engines can be found by placing its respective piston speed limit equal to $C_l\sqrt{L}$. Then by substituting the corresponding value of C_l as given in Table VII, (10) can be solved for the limiting stroke L.

The accompanying cylinder-bore limits for any particular kind of service can be determined by dividing such stroke limit by a suitable stroke-bore ratio. Table VIII has been determined on this basis and shows the limiting cylinder dimensions at which excessive heat-flow begins to exert a marked influence upon piston speed restriction. This table also specifies the assumed full-load values for the ratio of MEP/E for both two and four-stroke cycle engines, with which the rate of heat-flow and other given data have been estimated.

The check results as found for the limiting sizes of such engines are set forth in Table IX. The specified values for the cast iron cylinder wall thickness S_n are intended to represent average or normal practice and include a customary rebore allowance.

Bore Temperature Limits

By plotting some of the more important data given in Table IX, certain limiting design constants can be deduced for high-speed engines. Starting with the bore temperature t_b as fixed by (9) and plotted in Fig. 4, the steady rise of the bore temperature t_b with increased heat-flow shows this to be one of the principal factors in speed limitation. As found from this curve, the maximum allowable temperature that can still be expected to insure satisfactory piston lubrication is

$$t_b \text{ lim.} = 350 + H/10 \text{ deg. fahr.} \qquad (11)$$

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TABLE VII—COMPARATIVE PISTON SPEED LIMITS

No.	Kind of Engine	Four-Stroke Cycle			Two-Stroke Cycle		
		$PS=Cl\sqrt{L}$	*PS lim.* Ft./Min.	*L/D*	$PS=Cl\sqrt{L}$	*PS lim.* Ft./Min.	*L/D*
	GAS AND GASOLINE ENGINES						
A	S. A. stationary, horizontal	$160\sqrt{L}$	800	1¼–1½	$125\sqrt{L}$	600	1¼–1½
B	Double-acting stationary	.....	800	1¼–1½	.....	720	1¼–1½
C	Small marine, heavy duty	$200\sqrt{L}$	750	1¼–1½	$200\sqrt{L}$	500	1 –1½
D	Small marine, high duty	$300–500\sqrt{L}$	1500	1¼–1½	$250\sqrt{L}$	650	1 –1½
E	Small marine, racing	$1000–1200\sqrt{L}$	3000	1½–1⅝			
F	Airplane, direct-connected	$500–550\sqrt{L}$	1400	1¼–1½	$500\sqrt{L}$	1100	1
G	Airplane, gear reduction	$850\sqrt{L}$	2200	1½			
H	Motorcycle	$800\sqrt{L}$	1500	1½	$500\sqrt{L}$	700	⅝
I	Four-cyl. automobile, moderate speed	$550\sqrt{L}$	1500	1¼–1½			
J	Twelve-cyl. automobile, high speed	$1100\sqrt{L}$	2500	1⅝			
K	Truck and tractor engines	$450\sqrt{L}$	1100	1⅛			
	SINGLE-ACTING OIL ENGINES						
L	High compr., submarine	$250\sqrt{L}$	1100	1 –1¼	$250\sqrt{L}$	1000	1¼–1½
M	High compr., cargo boat	$150\sqrt{L}$	800	1¼–1⅜	$140\sqrt{L}$	750	1¼–1⅜
N	Low compr., marine				$190\sqrt{L}$	750	1¼–1½
O	High compr., horiz. stationary	$175\sqrt{L}$	900	1¼–1⅜	$175\sqrt{L}$	700	1¼–1½
P	Low compr., horis. stationary	$175\sqrt{L}$	900	1¼–1⅜	$175\sqrt{L}$	700	1¼–1⅜

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TABLE VIII—LIMITING CYLINDER DIMENSIONS FROM EQUATION 10

No.	Kind of Engine	Four-Stroke Cycle						Two-Stroke Cycle					
		$D \times L$, Inches	N, R.p.m.	MEP, Lb./ Sq. In.	Ind. Eff. E, per Cent	$\frac{MEP}{E}$	Jacket Loss J, per Cent	$D \times L$, Inches	N, R.p.m.	MEP, Lb./ Sq. In.	Ind. Eff. E, per Cent	$\frac{MEP}{E}$	Jacket Loss J, per Cent
	SINGLE-ACTING GAS AND GASOLINE ENGINES												
A	S. A. stationary, horizontal	19 x25	190	80	30	207	29.3	15 x23	155	65	28	232	21.9
B	Double-acting stationary	23 x30	160	75	30	250	27.1	19 x30	145	70	28	250	20.0
C	Small marine, heavy duty	10 x14	325	85	22	385	44.5	5¼x 6¼	480	45	18	250	39.0
D	Small marine, high duty	7½x 9	1000	80	22	364	41.3	6¼x 6¼	575	41	17	241	36.3
E	Small marine, racing	5 x 7½	2400	90	25	300	38.3						
F	Airplane, direct-connected	5½x 7	1200	100	25	400	42.9	5 x 5	1300	60	20	300	33.4
G	Airplane, gear reduction	4½x6¾	1950	100	25	400	41.9						
H	Motorcycle	3½x 3½	2500	75	20	375	49.0	1¾x 2	2100	40	16	250	46.5
I	Four-cyl. automobile, moderate speed	5½x 7½	1200	80	22	364	43.7						
J	Twelve-cyl. automobile, high speed	3½x 5	3000	75	20	375	46.4						
K	Truck and tractor engines	4½x 6	1100	80	22	364	47.4						
	SINGLE-ACTING OIL ENGINES												
L	High compr., submarine	17 x19	350	100	38	263	19.6	13 x16	375	95	36	264	15.7
M	High compr., cargo boat	19 x28	170	95	40	237	19.9	17 x28	160	90	38	237	15.4
N	Low compr., marine							13 x15	300	40	21	190	23.5
O	High compr., horiz. stationary	18 x26	210	90	40	225	19.8	12 x16	260	85	38	224	15.9
P	Low compr., horiz. stationary	18 x26	210	50	23.5	217	30.7	12 x16	260	40	21	190	23.3

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TABLE IX—HEAT-FLOW AND STRESS DATA FOR LIMITING ENGINE SIZES

No.	Four-Stroke Cycle							Two-Stroke Cycle						
	Surface Factor F	Heat-Flow H	Wall Thickness Sn, In.	Bore Temp. tb, Deg. F.	Wall Stress, Lb./Sq. In.			Surface Factor F	Heat-Flow H	Wall Thickness Sn, In.	Bore Temp., tb, Deg. F.	Wall Stress, Lb./Sq. In.		
					Press., fp	Temp., ft	Total, $fp+ft$					Press., fp	Temp., ft	Total, $fp+ft$
SINGLE ACTING GAS AND GASOLINE ENGINES														
A	2.68	276	1 11/16	358	2930	4240	7170	2.17	269	1	337	2770	3480	6250
B	3.19	252	1 1/2	370	2850	4880	7730	2.79	333	1 1/16	393	2775	5360	8135
C	1.63	505	5/8	406	2080	4880	6960	0.87	336	13/32	304	1700	1760	3460
D	1.14	942	13/32	459	2510	5730	8240	0.96	485	7/16	340	1970	2360	4330
E	0.81	1550	5/32	581	3320	5730	9050							
F	0.82	1010	1/4	488	3920	3280	7200	0.70	1060	1/4	449	2910	3440	6350
G	0.73	1420	9/32	514	3850	4000	7850							
H	0.47	1200	5/16	456	2190	2920	5110	0.28	685	5/16	352	950	1680	2630
I	0.88	930	11/32	448	2090	4120	6210							
J	0.53	1520	5/16	521	2190	3680	5870							
K	0.72	757	11/16	396	2080	3040	5120							
SINGLE-ACTING OIL ENGINES														
L	1.88	352	1 7/16	427	3050	6600	9650	1.42	498	1 1/8	463	3020	7200	10220
M	2.42	303	1 1/8	364	2540	4920	7460	2.16	283	1 11/16	407	2540	6170	8710
N								1.83	317	3/4	324	2060	3080	5140
O	2.28	215	1 11/16	369	2540	5040	7580	1.37	290	1 1/4	374	2440	4840	7280
P	2.88	239	1 1/4	333	1880	3480	5360	1.78	276	1 11/16	313	2060	2440	4500

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Fig. 4 shows a bore temperature of from 400 to 450 deg. fahr. to be a desirable limit. In fast running engines it is apparently necessary to work with an even hotter bore, but the total wall stress $f_p + f_t$ should not be allowed to exceed the prescribed limits.

In case of explosive engines, hot spots in the cylinder must also be guarded against; otherwise preignition is likely to result. While this restriction does not apply to oil engines with timed injection, the plotted bore temperatures for both two and four-stroke cycle oil engines are almost identical with those of explosive engines. The bore temperature can therefore be taken as the primary factor in fixing rotative speed limits, provided that the valves and head parts are kept under the requisite temperature control by adequate design measures.

Cylinder-Wall Thickness

The pressure stress f_p as derived from (8) for the limiting engine sizes in Table VIII is found to be approximately equal to

$$f_p = 2000 + PD/10 \qquad (12)$$

The normal cylinder-wall thickness S_n that applies for an explosive pressure of 300 lb. per sq. in. when figured upon the basis of (12), is plotted as curve *A* in Fig. 5. It is expedient to maintain the pressure stress f_p at the highest allowable value so as to work with the minimum

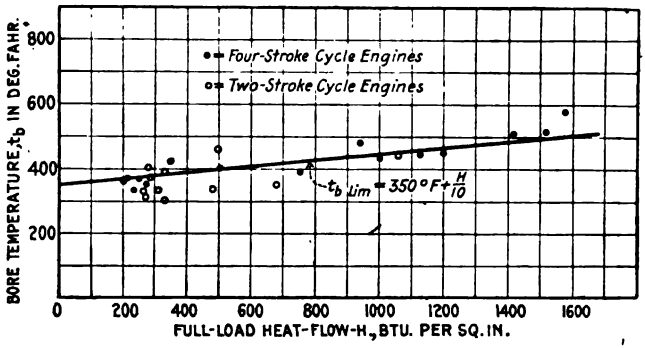

Fig. 4—Curve Showing Bore Temperature Limits

combined stress $f_b + f_t$. That the pressure stress specified in (12) can be increased to advantage is apparent from the following considerations:

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The minimum safe thickness for a simple thin cylinder as determined solely upon a pressure stress basis is $PD/2 \times 3750$, where the 3750 is the allowable pressure stress for a thin cast iron cylinder of 20,000 lb. per sq. in. ultimate strength at a gross factor of safety of $4 \times 2 \times 4/3$.

If to this a suitable but close rebore allowance be added, we have

$$S_{min} = \frac{PD}{7500} + \frac{\sqrt{D}}{20} \qquad (13)$$

where S_{min} = minimum allowable cylinder-wall thickness in inches.

This formula appears to give satisfactory values throughout a wide range of bore sizes, and in fast running engines can be expected to provide for the minimum total stress $f_p + f_t$. For hard close-grained cast iron, S_{min} can be reduced in proportion to its increased ultimate strength.

For the smaller sizes of slow running engines where no disturbing effects are set up by heat-flow, the use of a cylinder-wall thickness of about $1\frac{1}{4}S_{min}$ provides for stiffer construction, and facilitates the work of casting and machining the cylinders. Excessive bore allowance is not advisable in high-speed engines, as it tends to augment cracking troubles and thus may defeat the intended purpose of lengthening the cylinder life.

Values for S_{min} as determined from (13) for $P = 300$ lb. per sq. in., have been plotted in Fig. 5 as curve *B;* the average value of S_{min} is about $\frac{7}{8}S_n$, which shows that the customary wall thickness can be reduced with advantage, especially in larger engines.

Limiting Heat-Flow Constants

The allowable value for the heat-flow H as based upon the prescribed bore temperature limits can be found by equating (9) and (11) and transposing thus:

$$H_{lim} = \frac{427}{S + 0.133} = \frac{425}{S + \frac{1}{8}} \text{(approx.)} \qquad (14)$$

The heat-flow limit is inversely proportional to an apparent wall thickness $S + \frac{1}{8}$, the constant $\frac{1}{8}$ being an equivalent wall allowance required to drive the heat from the outer wall surface into the jacket water. It may be necessary to reduce further the given value of H_{lim} in case the metal lacks uniformity or when the head parts

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are made materially heavier than the bore wall. Large valve openings and massive cylinder flanges localize heat-flow, and this tends to increase temperature stress and imposes additional speed restriction.

A detailed discussion of this aspect would lead too far from the present subject to warrant treatment here. The aim of the present paper is rather to determine the limiting speeds that apply to an ideal engine having the minimum of wall thickness and unencumbered with valve openings of a size sufficient to dominate speed restriction.

Surface Factor Formulas

In engines in which the combustion chamber is simply formed in an extended portion of the cylinder bore, the

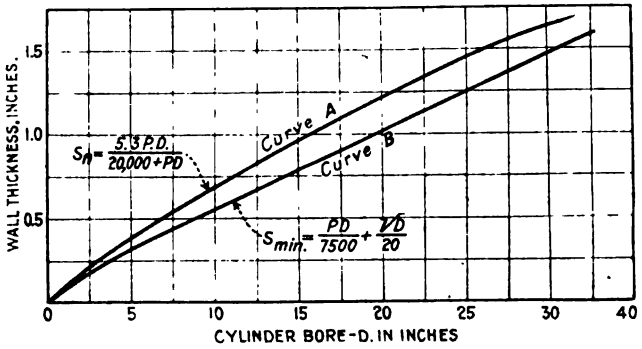

Fig. 5—Normal (Curve A) and Minimum (Curve B) Cylinder Wall Thickness for 300 Lb. per Sq. In. Explosive Pressure

normal surface factor assumes the following numerical value when the entire piston area is taken as effective cooling surface:

$$F_n = \frac{D}{4}\left(\frac{\frac{l_o}{L} + \frac{1}{4}}{\frac{l_o}{L} + \frac{1}{4} + \frac{1}{2r}}\right) \qquad (15)$$

where $F_n = V_o/A_o$ = normal surface factor for the specified form of combustion chamber.

l_o/L = ratio of clearance volume to piston displacement, as measured in terms of the stroke L.

$r = L/D$ = stroke-bore ratio.

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Surface factors applying to combustion chambers with more intricate contour can be approximated by multiplying the given value of F_n by a suitable constant. In any case, the surface factor can be expected to increase directly with the bore diameter D. The usual values for F_n as determined for a wide range of conditions are plotted in Fig. 6. High compression oil engines possess a relatively large surface per unit of volume. For the more common low compression explosive engines, with r varying from 4/3 to 3/2 at a clearance volume of 20 to 25 per cent piston displacement, the mean value of F_n is generally found to be about $D/7.2$, as indicated in Fig. 6.

Limiting Speed Formulas for Low Compression Engines

Using the average value of $F_n = D/7.2$ as a basis for the further discussion and placing (14) equal to (1), the following relation is obtained:

$$\left(\begin{matrix}\text{For four-stroke}\\ \text{cycle engines}\end{matrix}\right)\left(\frac{N}{\log_{10}N}\right)_{lim.} = \frac{1{,}600{,}000}{(S+\tfrac{1}{8})\left(\frac{MEP}{E}\right)^{\frac{1}{3}}D^{\frac{2}{3}}} \tag{16}$$

$$\left(\begin{matrix}\text{For two-stroke}\\ \text{cycle engines}\end{matrix}\right) = \frac{1{,}000{,}000}{(S+\tfrac{1}{8})\left(\frac{MEP}{E}\right)^{\frac{1}{3}}D^{\frac{2}{3}}} \tag{16a}$$

The above equations permit finding the maximum allowable speed at which any given size of explosive gas or oil engine reaches the prescribed bore-temperature limit. Fig. 7 shows the relation of $N/\log N$ as plotted upon a speed basis; its numerical value can be closely approximated by the curve $1.07N^{\frac{5}{6}}$ as indicated.

Equation (16) can be further simplified by confining the cylinder wall to the minimum allowable thickness S_{min} as defined by (13), in which case the value of the factor $(S + \frac{1}{8})$ at 300-lb. explosive pressure can be taken as proportional to the cylinder bore, thus

$$(S_{min} + \tfrac{1}{8}) = 0.15D^{\frac{2}{3}} \text{ (approx.)} \tag{17}$$

With this further restriction the above equations, as confined to cast-iron cylinders working with F_n = about $D/7.2$ and at an explosive pressure of about 300-lb. per sq. in., finally take the form,

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$$\left(\begin{matrix}\text{For four-stroke}\\ \text{cycle engines}\end{matrix}\right)\left(\frac{N}{\log_{10}N}\right)_{lim} = \frac{10{,}700{,}000}{\left(\frac{MEP}{E}\right)^{1/3}D^{1/3}} \tag{18}$$

$$\left(\begin{matrix}\text{For two-stroke}\\ \text{cycle engines}\end{matrix}\right) = \frac{6{,}700{,}000}{\left(\frac{MEP}{E}\right)^{1/3}D^{1/3}} \tag{18a}$$

Corresponding values for piston speed limits are:

$$\left(\begin{matrix}\text{For four-stroke}\\ \text{cycle engines}\end{matrix}\right)PS_{lim} = \frac{DrN}{6} = \frac{1{,}800{,}000\, r\log_{10}N}{\left(\frac{MEP}{E}\right)^{1/3}D^{1/3}} \tag{19}$$

$$\left(\begin{matrix}\text{For two-stroke}\\ \text{cycle engines}\end{matrix}\right) = \frac{1{,}100{,}000\, r\log_{10}N}{\left(\frac{MEP}{E}\right)^{1/3}D^{1/3}} \tag{19a}$$

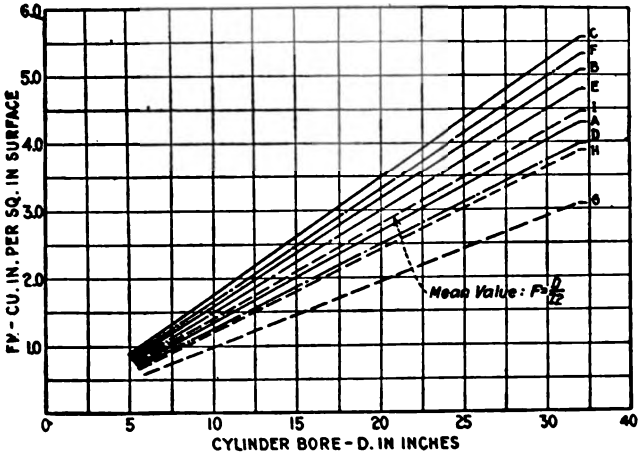

FIG. 6—CURVES SHOWING NORMAL SURFACE FACTORS (F_n) FOR DIFFERENT CYLINDER FORMS

Curve	*A*:	$l/L = 1/2$ when $L/D =$	1.0
	B:		1.5
	C:		2.0
	D:	$l/L = 1/4$ when $L/D =$	1.0
	E:		1.5
	F:		2.0
	G:	$l/L = 1/16$ when $L/D =$	1.0
	H:		1.5
	I:		2.0

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The curves in Fig. 8, which are derived from (18) and (19), show the manner in which heat-flow limits rotative speed in four-stroke cycle explosive engines. The restrictions imposed by heat-flow are especially marked in sizes above 10-in. bore; the allowable piston speed for large four-stroke cycle engines working with a relatively high input per cubic foot of displacement, drops to a low value as shown by curve *C*. No material advantage is gained by striving for too high a value of *MEP/E* in single-cylinder high-powered engines, because the difficulties accompanying increased heat flow necessitates a reduction in rotative speed.

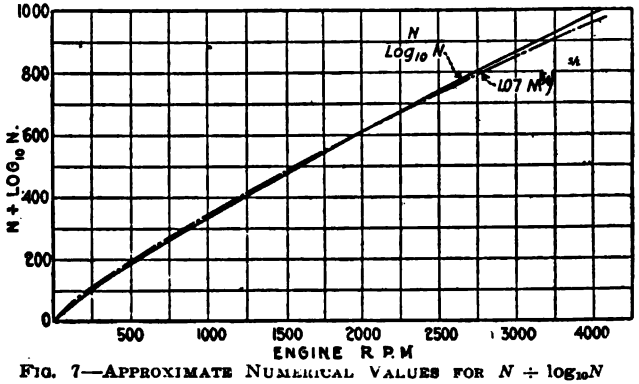

FIG. 7—APPROXIMATE NUMERICAL VALUES FOR $N \div \log_{10} N$

When practicable, this restriction can be overcome by casting the cylinder-liner and head parts of manganese bronze and like metals, since then the speed can be increased in proportion to the superior heat conductivity of such metals.

As applied to small four-stroke cycle engines, the curves in Fig. 8 show that cast iron cylinders, when properly designed for heat-flow, should readily be able to meet the rather severe requirements of modern high-speed automobile and marine engines. This deduction is dependent upon uniform and effective cooling for all cylinder parts and upon a suitable shape for the head wall, so that it can withstand the explosive pressure without exceeding the bore wall-thickness S_{min}.

The speed constant that will prove most suitable for any particular kind of engine is naturally dependent upon the service for which it is intended. As deduced in round figures for the various kinds of four-stroke cycle gas and gasoline engines listed in Table VII, the following multiplying factors for the stipulated speed constant 10,700,000 (18) represent current practice for cast iron cylinders:

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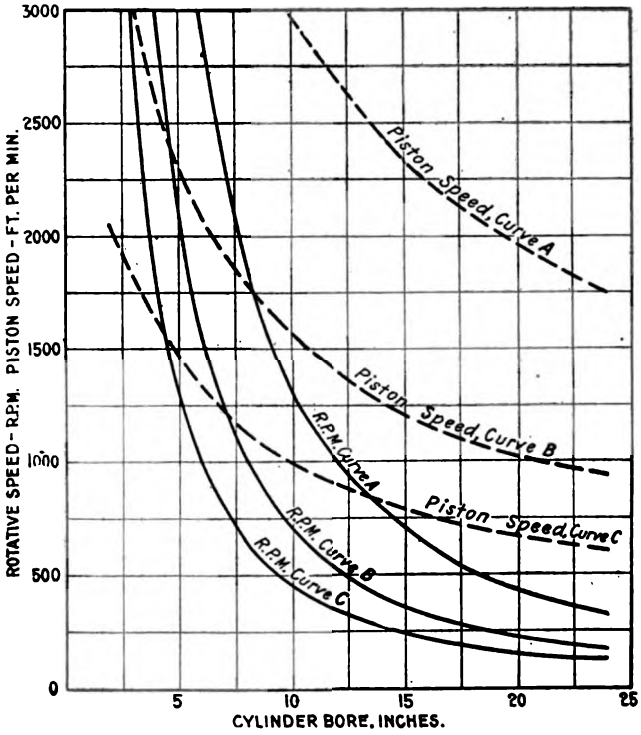

FIG. 8—CURVES SHOWING EFFECT OF HEAT-FLOW UPON PISTON AND ROTATIVE SPEEDS OF 4-STROKE CYCLE ENGINES

Curve *A* for $MEP/E = 200$
Curve *B* for $MEP/E = 300$
Curve *C* for $MEP/E = 400$
} $L/D = 4/3$

TABLE X—MULTIPLYING FACTORS FOR SPEED-CONSTANT

Reference No.	Kind of Engine	Multiplying Factor
A, B & C	Stationary and heavy-duty marine	About ½ to ¾
D	High-duty marine	1 to 1¼
E	Racing engine	1½
F	Airplane, direct-connected	1
G	Airplane, gear reduction	1¼
H	Motorcycle	¾
I	Four-cyl. automobile, moderate speed	¾
J	Twelve-cyl. automobile, high speed	1
K	Truck and tractor engines	¾

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The safe speed limit is largely fixed by the character of service required; when continuous full-load service is demanded, it is essential to reduce the given piston speed limit PS_{min} in proportion to the given multiplying factors, so as to provide a sufficient margin of safety against excessive wear and breakdown.

While most four-stroke cycle engines operate at between 75 and 100 per cent of the so-prescribed speed limits, the corresponding average for two-stroke cycle engines is considerably lower, particularly as regards the small crankcase compression gasoline engines, which only reach about 50 per cent of the specified limit values. The low speed efficiency of most engines of this type is to be attributed in large part to excessive bore temperature, a matter that could no doubt be considerably improved by the use of thinner cylinder walls and by giving more careful attention to heat-flow effects in the design of two-stroke cycle engines.

Two Versus Four-Stroke Cycle Engines

The various formulas show that when working with high mean effective pressures the two-stroke cycle engine is not so well adapted for fast running as is the four-stroke cycle. The two-stroke cycle can be used to best advantage when confined to long-stroke slower-running gas engines working with a relatively lower heat input per cubic foot of displacement. The lower first cost resulting from simplicity of construction and the elimination of mechanically operated valves does however give the two-stroke cycle engine a considerable commercial advantage for certain kinds of service where slow running is essential.

In the case of oil engines, the conditions become more

Table XI—Comparative Properties of Diesel Submarine Engines

Item	Four-Stroke Cycle	Two-Stroke Cycle
Indicated efficiency (E), per cent	40	38
MEP, lb. per sq. in.	100	95
Cylinder dimensions, inches	15 by 15½	12 by 13½
Rev. per min. (N)	425	450
Surface factor F, about	1.5	1.37
Heat flow H by (1), B.t.u. per sq. in.	337	450
Cylinder thickness (S_{min}), inches	1.2	1.6
Bore temperature t_b by (9), deg. fahr.	390	420
t_b *lim.* by (11), deg. fahr.	384	395
Temp. stress f_t by (7), lb. per sq. in.	5220	5810
Press. stress f_p by (8), lb. per sq. in.	3250	3110
Total wall stress f_t+f_p, lb. per sq. in.	8470	8920

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favorable to the two-stroke cycle. The following comparison is made between the two and four-stroke cycle Diesel engines commonly used for submarine drive. Both engines are assumed to have an approximately equal rated power capacity of about 110 b.hp. per cylinder. Table XI presents the data used in arriving at a heat-flow comparison applying to these two types of single-acting Diesel engines.

The four-stroke cycle Diesel engine offers no conspicuous advantage as regards heat-flow when compared upon the basis of equal cylinder-power capacity. The two-stroke cycle engine, because of its smaller cylinder dimensions, can take care of its inherently greater heat-flow without destructive temperature effects. Furthermore, while the wall stress of the two-stroke cycle engine is shown to be greater by some 5 per cent, the conditions in the case of a high-speed four-stroke cycle engine are not really so favorable as the estimated stress relations would indicate. The large openings required for the four-stroke inlet and exhaust valves tend to weaken the cylinder-head structure and do not allow of the same uniform distribution of metal and effective cooling of the head-wall as in the case of a two-stroke cylinder when charged from below by means of piston-controlled inlet ports.

Both types of submarine engines slightly exceed the prescribed bore-temperature limits. A reduction of at least 10 per cent in the given speed would provide for greater reliability of operation, especially so since dependable service is absolutely essential for a successful submarine engine.

Jacket Loss Formulas

The full load jacket loss of any internal combustion engine, as measured in terms of heat input, is readily deduced from the foregoing equations. As based upon (2), the heat input of a four-stroke cycle engine, determined for a period of one hour, is

$$Q = 0.185 \frac{MEP}{E} \times \frac{N}{2} \times 0.60 \frac{V_o}{1728} = 0.0032 N \frac{MEP}{E} V_o \tag{20}$$

The jacket loss occurring during this same period is measured by the product of the effective area A_e and the heat-flow rate H, and when taken with respect to the heat input becomes

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$$J = \frac{A_c H}{0.0032\, N \frac{MEP}{E}\, V_o}$$

Where J = full-load jacket-loss as measured in terms of heat input,

$V_o = \pi/4D^2L$ = piston displacement in cubic inches.

Substituting for the value of H as given in (1), the formula for jacket-loss finally takes the form:

$$\left(\begin{matrix}\text{For four-stroke}\\ \text{cycle engines}\end{matrix}\right) J = \frac{\left(\frac{l_o}{L} + \frac{1}{4}\right)\left(\frac{MEP}{E}\right)^{\frac{1}{8}}}{3.2\, F^{\frac{1}{8}} \log_{10} N} \qquad (21)$$

The relative jacket-loss for two-stroke cycle engines is found to be somewhat lower, and can be determined by substituting 3.9 in place of the constant 3.2 given in the above equation.

According to (21), the full load jacket-loss J increases directly with the temperature factor, while increased speed and a large volume content per unit surface tend to reduce the jacket loss percentage. The speed and surface factors N and F are largely predetermined by considerations other than their effect upon jacket loss, but as checked from Table IX the product $\log_{10} N \times F^{\frac{1}{8}}$ can be expected to lie between the limits 2¾ and 3¼, with approximately 3.0 as a fair average value.

The estimated jacket loss applying to the various kinds of engines previously analyzed for speed limits are presented in Table VIII. This estimate shows the jacket loss in the larger high-compression engines to be relatively small, that is, from 20 to 25 per cent of the heat input, while for the smaller low-compression high-speed engines the jacket loss may be about twice as large.

Constancy of Indicated Efficiency with Increased Jacket Loss

The jacket loss exerts no very marked influence upon the indicated thermal efficiency, largely because E is dependent primarily upon the ratio of expansion, as is evident from the following formula:

$$E = 1 - 1/R^{m-1} \qquad (22)$$

where $R = \frac{L + l_o}{l_o}$ = ratio of expansion

m = exponent of expansion line PV^m = Const.

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For most explosive engines working with a reasonably clean and quick-firing charge, the average value of m is about 6/5. The value of m does not change materially with change in bore dimensions. Since the thermal efficiency of an explosive engine is dependent solely upon the difference of the initial and final temperatures of expansion, any additional jacket cooling will tend to reduce the exhaust loss by an amount practically equal to the increase of jacket loss, but without producing any material loss of indicated efficiency.

In fast running engines a loss of from 5 to 10 per cent in thermal efficiency can be expected to result from the sloping explosion line shown in Fig. 9. The relatively slow rate of combustion reduces the effective ratio of expansion in approximately the manner indicated.

Importance of Small Surface Factors

The preceding formulas have shown the important function of the surface factor F and the advantage of providing ample cooling surface for the heat offtake. Skillful manipulation of this factor should lead to further improvement in internal-combustion engine design, especially so for the larger explosive engines, in which the cooling surface is generally inadequate. In the past, excess heat-flow has been the bugbear of many of the larger high-speed engines, since it frequently leads to preignition troubles, cracking of cylinder parts and other equally unsatisfactory operative results.

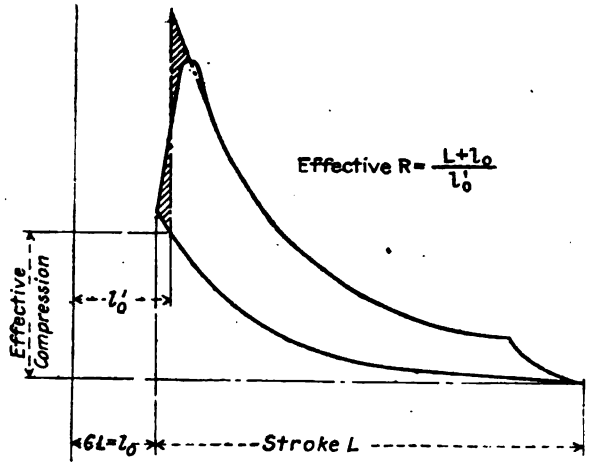

Fig. 9—Indicator Card Illustrating Effective Expansion Ratio in High-Speed Engine

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For high-powered internal-combustion engines to compete successfully with other forms of modern prime movers, high speed is today essential. In attempting to effect a speed increase by the use of more ample cooling surface, care must be taken to keep the combustion chamber reasonably free from pockets which are likely to trap the products of combustion and so prevent the formation of a fairly uniform and quick-firing explosive mixture throughout the working charge.

Experience with small high-speed engines shows conclusively that when considered apart from the limitation imposed by heat-flow, piston speeds as such can be increased far in excess of the limits now commonly set for the larger sizes of internal combustion engines. It is not improbable therefore that the future will see engines of large dimensions run at piston speeds that have heretofore been deemed prohibitive. To make any substantial advance along these lines will require careful attention to design details, especially so as regards heat-flow effects and lubrication.

Author's Conclusion

On the basis of the principles enunciated, it may be concluded that the full-load heat-flow through gas and oil engine cylinders is dependent upon three prime factors as expressed in equation (1), as follows:

(1) Temperature factor, $\left(\frac{MEP}{E}\right)$

(2) Speed factor, $N/\log_{10}N$.

(3) Surface factor F, effective volume per unit of cooling surface.

The designer usually strives to keep the first and second of these factors at a maximum in order to secure a correspondingly large engine output. The resulting high rate of heat-flow tends unduly to raise the temperature of the cylinder bore and head walls. The permissible rate of heat-flow is shown to be strictly limited by the resulting bore temperature, which should not exceed 400 to 450 deg. fahr., as fixed by the given method of temperature determination.

For a given rate of heat-flow, the temperature drop increases directly with increased thickness of the cylinder wall. Hence the maintenance of the same bore temperature in all sizes of engines necessitates a reduction in heat-flow with increased bore dimensions. Large

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engines are shown to reach the prescribed heat-flow limit at comparatively slow rotative speeds.

This inherent restriction can be overcome in part by making the cylinder liners of manganese bronze and like metals having a far higher thermal conductive power than is possessed by cast iron. In larger engines such innovation can be expected to lead to prohibitive first costs; hence, both from constructive as well as from commercial aspects, the author advocates the skillful manipulation of the surface factor F as the most promising method of overcoming the destructive heat-flow effects in the cylinder and head parts of large internal combustion engines.

The numerical value of the surface factor F normally changes directly with the bore dimensions, and for large engines the available cooling surface per unit of effective volume is comparatively small. On the other hand, a relatively large cooling surface per unit of volume content, as used in the case of smaller engines, serves better to diffuse the jacket loss and prevent excessive concentration of heat-flow through the cylinder wall. This results in a considerable improvement in the temperature relations and tends to eliminate preignition, excessive heating of the valves and like temperature troubles. To prevent cracking of the cylinder parts, the sum of the temperature and pressure stresses should not exceed prescribed limits.

While the proposed artificial increase in the cooling surface does increase the jacket loss, it does not materially affect the indicated efficiency. It may be expected, therefore, to improve operating conditions generally when applied to large high-speed engines.

Formulas are derived to fix the maximum allowable speed for both two and four-stroke cycle gas and oil engines as based upon heat-flow effects. For usual values of the surface factors F and the prescribed minimum cylinder wall thicknesses, the piston speed limits are shown by (19) to be directly proportional to the stroke-bore ratio and inversely proportional to the product of the temperature factor and the cube root of the bore diameter. Hence, excessive heat-flow may restrict piston speed even in the smaller high-speed engines when working with relatively large values of MEP/E.

The inverse ratio of the factor D further shows that under similar conditions of heat-flow it is not safe to run the piston speed in larger engines as fast as in smaller engines, unless compensated for by a corresponding increase in the stroke-bore ratio.

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The rate of heat-flow in two-stroke cycle engines can be taken at about 1.6 that of the four-stroke cycle engine for equal numerical values of the three primary factors previously referred to. A comparison with Diesel submarine engines of equal power capacity per cylinder shows that as applied to this particular service the four-stroke cycle offers no marked advantage as viewed from the heat-flow standpoint.

The four-stroke cycle does, however, offer an important advantage when working with high values of MEP/E, such as apply in fast running marine, automobile, airplane and like gas engines. Under this condition the restriction imposed by the increased rate of heat-flow would keep the rotative speed of two-stroke cycle engines below commercial requirements.

The speed limitation imposed upon practically all kinds and sizes of internal-combustion engines, both of the two and four-stroke cycle types, are shown to be identical, all being dependent primarily upon the allowable rate of heat-flow set up through the cylinder wall. In cases where well-designed engines are not speeded up to the prescribed limit such speed reduction is usually found to conform with the intended service and the general shop practice of the engine maker.

Thus the interdependence of interest between the kindred branches of internal-combustion engineering becomes evident, and the need is shown of keeping in reasonably close touch with the progress made in the engine industry considered as a whole.

Finally, attention is directed to the necessity of further experimental data relating to jacket-loss tests, and it is hoped that some one of our well-equipped university laboratories may see fit to undertake the extensive research investigations required to establish fully the underlying laws of heat-flow outlined in this paper.

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CHICAGO TRACTOR MEETING PAPERS

FUNDAMENTALS OF TRACTOR DESIGN

By George T Strite*

The fundamental design of a tractor involves, first of all, the size of the tractor, the number of plows it will pull, or the drawbar pull. For many years the tendency has been to keep increasing the size just a little. Even in the days of the big tractor, eight and ten-plow, I can remember when the influence exerted by our customers for just a little more power was hard to resist. The temptation was continually before us to build something larger, although at that time most of us knew that eventually the small tractor was going to be the thing.

Since the complete readjustment of the tractor manufacturing business, most companies have started with the two and the three-plow outfit, and ever since there has been a constant demand for just a little more pulling capacity. One of the hardest things in the tractor business is to have the nerve to "stand pat" and build one size and not keep constantly increasing the size so that it will do a little more work.

Power and Traction

After the size has been determined, we have two factors to deal with—one, the *amount of power;* the other, *traction.* Here again is a hard question to decide, whether to have sufficient power so that the engine will not stop and will continually turn the wheels regardless of the kind of ground or footing; or, on the other hand, to install an engine with only sufficient power to carry the load and not slip the wheels. If an engine develops five or ten horsepower more than would be required for the ordinary work, then the entire transmission and tractor must have sufficient strength to take care of the load and turn the wheels under extreme conditions. Would we think of buying an automobile today with engine, transmission and bearings of such capacity as to spin the wheels on high gear and dry ground? This would be absurd.

However, from a sales point of view, there is one reason

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why we have done this in the case of the tractor. In demonstrating the tractor, at which time it is generally loaded to the limit, if we stall the engine, it is most discreditable to the tractor; if the engine has sufficient power to keep on turning the wheels, even to the extent of burying the tractor in the ground, this brings out applause for the tractor, and the farmers will say: "Gee, but that tractor has got the power!"

Personally, I think that we have carried this too far. What the average wheel tractor needs is more traction as compared with the amount of power furnished. The farmer would then get a more efficient and better balanced machine, a machine that could be used in extremely bad conditions, on wet, soft or on sandy ground. As soon as the farmer is educated on this point, we will begin to build tractors better balanced for total average efficiency, and not those that will do certain tricks, regardless of the average work they are expected to do. The farmer or manufacturer will then use more horse sense and good judgment with an eye to final average efficiency.

When we have settled the ratios between power and traction, the third item to consider in the design of a tractor is the bearing surface of the wheels on the ground as compared with the power of the tractor. In the old days we built tractors which were exceedingly heavy and deficient in surface. The tractor of today is better, but the future tractor will be built with less weight because of multiple-cylinder engines, high-grade material, and greater wheel surface on the ground. The wheel surface must also be backed up by the proper shape and style of grouters or lugs.

Weight Distribution

After the total weight of the tractor is determined, then comes the question of distribution. If it is a four-wheel machine with two drivers in the rear and two front wheels, careful judgment must be used as to how much weight should be carried on the drive wheels and how much on the front wheels. With the best practice today about 70 per cent of the total weight seems to be carried on the drive wheels and about 30 per cent on the front wheels. This weight must be calculated with the tractor standing still, without any drawbar load.

When the tractor is working with a reasonably heavy drawbar pull, a large percentage of the total weight plus a certain percentage of the drawbar pull will be

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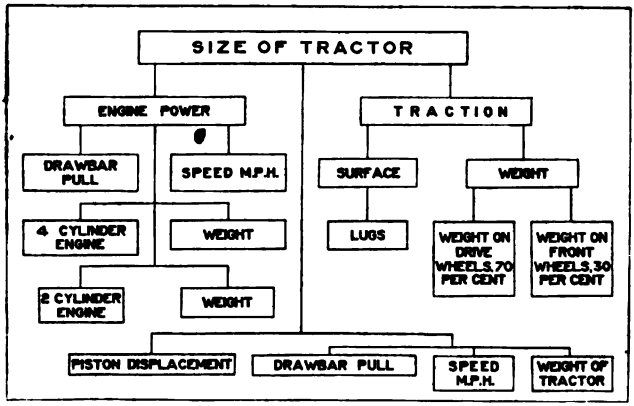
carried on the rear wheels. This percentage depends upon the height of the drawbar hitch on the tractor, the height of the hitch on the plow, and the load being pulled. From 10 to 15 per cent of the weight is thus carried on the front wheels, which for all practical purposes is enough to steer the tractor, except when going up a steep grade, when the front wheels are apt to rise and make hard steering.

The power required of the tractor engine depends, first, upon the drawbar pull, which, however, has no significance unless the speed is considered. It takes practically the same power to pull two plows at three miles per hour as to pull three plows at two miles per hour.



FACTORS AFFECTING TRACTOR DESIGN

This is a point on which the farmer has been greatly deceived. Some companies have sold their tractors on the basis of drawbar pull in pounds, saying nothing about the speed in miles per hour, while other companies have made a strong point of the speed. Both, of course, must be considered in determining the drawbar horsepower.

From an engineering point of view, and I think eventually for the convenience of the farmer, the power should be specified in figures of total displacement. In the figures in the accompanying table the speed of the different tractors will not be considered. In group A there is little difference in the speed of the three machines. In group B in actual practical farm work there is surprisingly little difference in the speed, although there is a difference between group A and groups B and C, as the speed of the small tractors will average from

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10 to 15 per cent higher. However, for the sake of comparison, we will disregard the speed.

Out of 95 different models in the tractor specification books, I have taken 10 of the best-known makes, of which 500 or more are actually in the field.

In taking the ratings of the ten different makes of tractors, it was a hard matter to get all the ratings. Some rate by number of plows, others by belt power and drawbar horsepower, and I found it was especially hard to get some of the companies to state specifically their drawbar pull in pounds. Many companies do not publish this.

I do not claim that these figures are correct or accurate, but they are the nearest I could get. They tend to show the great variation in rating. Some tractors are very much over-rated and others conservatively rated. The main thing I wish to bring out is these variations, and the general average trend of the tractor in size, power in proportion to weight, etc., and also the marked refinement, development and efficiency in proportion to weight and piston displacement.

I think that tractor engineers who are designing new tractors should consider the average, past and present, in designing new machines and benefit by past experience.

The first three in group A are of the large type, two with two cylinder and one with a four-cylinder engine. Group B represents the four-cylinder, three and four-plow tractors of late type and light weight. In group C we have three of the two-cylinder, three and four-plow tractors of recent but possibly not of the latest design.

For comparison, I will take up: First, the piston displacement in cubic inches per minute, for each 14-in. plow pulled.

Second, the piston displacement, for each rated 100-lb. drawbar pull.

Third, the piston displacement for each rated drawbar horsepower.

Fourth, the weight of tractor for each 14-in. plow pulled.

Fifth, the weight of tractor for each 100-lb. drawbar pull.

Sixth, the weight of tractor for each rated drawbar horsepower.

In Col. 9, for example, tractor A-1 has a piston displacement of 147,250 cu. in. per min. for each 14-in. plow pulled. Tractor A-2 has a piston displacement of 86,206 cu. in. per min. for each plow pulled. Tractor

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A-3 has a piston displacement of 88,312 cu. in. per min for each plow pulled.

In Col. 9 the average piston displacement of the three tractors in group A is 107,256 cu. in. per min. per plow.

The average piston displacement per minute per plow of the four tractors in group B is 91,540, or about 15,000 cu. in. less than in group A.

The average piston displacement per minute per plow of the three tractors in group C is 85,771, or 21,000 cu. in. less than in group A. The two-cylinder tractors in group C have less displacement per minute, or less power per plow, than have those of group B, the four-cylinder machines of the same size.

In Col. 10, group A, the average piston displacement per 100 lb. drawbar pull is 14,581 cu. in. per minute. In group B this displacement is 11,247 cu. in., or about 3300 cu. in. less than the corresponding figure for group A. In group C the displacement is 12,164 cu. in., so that the displacement is 900 cu. in. more than that in group B, and 2400 cu. in. less than that in group A per 100 lb. drawbar pull.

In Col. 11, group A, the average piston displacement for each rated drawbar horsepower is 25,846 cu. in. per minute. In group B, the displacement is 23,544 cu. in. per minute. The displacement is 22,728 cu. in. in group C, so that the four-cylinder machines in group B have 700 cu. in. more displacement.

In Col. 13 the weights of the three different *groups of tractors* are compared. The average weight of the three large tractors in group A is 21,866 lb.; the average weight of the four-cylinder tractors in group B is 5150 lb.; and the average weight of those in group C, the two-cylinder tractors, is 5800 lb. or 650 lb. more per tractor than the four-cylinder machine. The average drawbar pull of group B tractors is 429 lb. more than that of the group C two-cylinder machine.

In Col. 14 are compared the weights for each 14-in. plow pulled. The average weight of the large tractor in group A is 2733 lb. for each 14-in. plow pulled.

The average weight of the group B tractors, the four-cylinder three and four-plow machines, is 1593 lb., or nearly 1200 lb. less than was the case with the group A tractors.

The average weight in group C two-cylinder tractors is 1815 lb. for each 14-in. plow pulled, or nearly 250 lb. more than for the average four-cylinder machine, and about 1100 lb. less than for the group A tractors.

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In Col. 15 we come to the comparative weight of the tractor for each rated drawbar horsepower. In the large tractor the average tractor weight is 681 lb. for each rated drawbar horsepower. In group B the four-cylinder tractor's average weight is 374 lb.; and in group C the tractor weight for the two-cylinder tractor is 482 lb. for each drawbar horsepower, or 108 lb. more than for group B and 200 lb. less than for the group A tractors.

Col. 16 gives the comparative weight for each 100 lb. of rated drawbar pull. In group A this is 374 lb.; in group B 199 lb., or 75 lb. less than was the case for group A; in group C the average is 258 lb. of tractor weight for each 100 lb. of drawbar pull, or 58 lb. more than for group B and 117 lb. less than for group A.

In these three groups of tractors the highest rate of piston travel is in group A and the highest revolutions per minute are in groups B and C.

Both the four-cylinder and two-cylinder late type tractor show a wonderful development, especially in regard to the weight per plow and per horsepower. The large tractor should weigh less for each plow and for 100 lb. of drawbar pull than the small tractor because in a one or two-plow tractor the initial weight of the tractor itself is great, as it is distributed over only one of two plows. The more plows we have to take this weight, the less the weight for each plow.

For instance, it takes an initial weight of from 1500 to 2000 lb. to pull one 14-in. plow, from 500 to 1000 lb. more to pull two plows, and from 300 to 700 lb. additional to pull the third plow. This also affects the power or displacement of the machine itself.

There is some variation in these average figures, which I think is caused by the fact that some tractors are overrated in drawbar horsepower and drawbar pull, while others are conservatively rated. As a rule, the four-cylinder machine is in a higher state of development than the two-cylinder machine. In one case the rated drawbar pull is almost equal to the entire weight.

There is a question as to how far we can go in reducing the weight per 100 lb. of drawbar pull for practical everyday use on the average conditions of soil. I believe we can eventually build tractors so that they will pull more than their weight, provided the speed is not too high, and that they are, as it were, geared to the ground, when used in certain sections where this is possible. I think this is not possible for average conditions.

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CHARACTERISTICS OF TEN WELL-KNOWN MAKES OF TRACTORS

Tractor Groupe	Horsepower Drawbar	Horsepower Brake	Drawbar Pull, Lb.	No. of Plows	No. of Cylinders	Cylinder Dimensions, In. Bore	Cylinder Dimensions, In. Stroke	Engine, R.P.M.	Cubic Inches Piston Displacement Per Rev.	Cubic Inches Piston Displacement Total per Minute	Per Plow	Per 100 Lb. Drawbar Pull	Per Drawbar Hp.	Piston Travel per Minute in Ft.	Pounds Weight Of Tractor	Per Plow	Per Drawbar Hp.	Per 100 Lb. Drawbar Pull
	1		2	3	4	5		6	7	8	9	10	11	12	13	14	15	16
A-1. ...	38-60		6,000	8	2	10x15		500	2,356	1,178,000	147,250	19,633	31,000	1,250	20,100	2,512	529	335
A-2	30-60		6,000	8	4	6½x8		650	1,061	689,650	86,206	11,494	22,988	866	19,500	2,437	650	325
A-3....	30-60		5,600	8	2	10x12		375	1,884	706,500	88,312	12,616	23,550	750	26,000	3,250	866	464
Averages.	.....		5,866	.....	... .	.....		.....	.....	.....	107,256	14,581	25,846	955	21,866	2,732	681	374
B-1 ...	12-20		2,250	3	4	4½x5		800	318	254,400	84,800	11,563	21,200	666	6,500	2,166	542	295
B-2.. .	18-36		3,300	3	4	4½x6¾		850	429	364,650	91,162	11,050	20,258	956	5,900	1,475	327	178
B-3. ...	12-25		2,500	3	4	4½x5½		900	312	280,800	93,600	11,232	33,400	825	5,200	1,733	433	208
B-4.. ..	15-30		2,600	3	4	4½x5¾		900	322	289,800	96,600	11,146	19,320	862	3,000	1,000	200	115
Averages..	.....		2,661	.....	.....	.....		.....	.....	.....	91,540	11,247	23,544	827	5,156	1,593	375	199
C-1.....	12-24		2,200	3	2	6x7		750	395	296,250	98,750	13,465	24,687	875	5,000	1,666	416	227
C-2.....	12-24		2,000	3	2	5½x7		750	332	249,000	83,000	12,450	20,750	875	5,000	1,666	416	250
C-3.....	12-25		2,500	3 or 4	2	6½x7		570	464	264,480	75,565	10,379	22,040	665	7,400	2,114	616	296
Averages..	.....		2,233	.....	.....	.....		.....	....	.....	85,771	12,164	22,738	802	5,800	1,815	482	257

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The Future of the Tractor

There is a big field in certain sections and for certain uses for the track-laying machine, but the wheel machine will be the popular machine and the big seller for average agricultural purposes.

Tractor engineers have been severely criticized in some cases for the past and present state of the tractor, with its different designs.

From now on, the tractor industry will advance either through natural evolution, the right tractors succeeding and the wrong gradually dying out, or by the tractor engineers and manufacturers getting together and *through standardization* avoiding failures that otherwise must come sooner or later.

The tractor manufacturers of today have the greatest opportunity that any manufacturing industry has ever had in the history of the world. They have an opportunity to carry standardization a long way successfully and with little resistance, if they will get together, work together, and through the Society of Automotive Engineers, assist in adopting the necessary standards.

It has been said that the automobile manufacturers will try to enter the tractor industry. They are going to get in, and personally I believe that the sooner they get in the better, although they will probably spend fortunes to learn just what many of us have alreadv learned. The automobile manufacturer has four great advantages over the tractor manufacturer.

First, he has the money, and is looked up to by the investor as a successful business man.

Second, he knows how to get production, and he has the organization and equipment for it.

Third, by past experience with the automobile he knows the value of standardization, and will help to standardize the parts of tractors.

Fourth, he has learned to cut out petty ideas, and to work for average results.

The tractor business is an important one. It is going to be one of the greatest in volume of any one business in the world. The tractor means more to the general happiness and welfare of mankind than any other machine that has ever been brought out. It is going to entirely change the farm power of all the world, and, further, it will revolutionize farming itself; better methods will mean more food for the world at far less cost.

Considering the state of the tractor industry at this

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time, the inability of the companies to produce tractors fast enough, and the quality of the tractors turned out, I believe that the Government should design and build a "Liberty" tractor to be manufactured by it during the war and to be open to any company to make, or to adopt any part of, after the war. The corps of engineers that design and build this machine should have instructions to do no experimenting, but to follow the lines of the best in the tractor as it is today.

This move would do more to bring about standardization and the eventual tractor than any other that could possibly be made. It will result in great good to all and especially to the tractor manufacturer. The war has already demonstrated the necessity and advantages of standardization of other machines. Why not the tractor?

THE DISCUSSION

H. L. HORNING (M. S. A. E.):—It seemed to be the consensus of opinion of the members of the S. A. E. in Washington that, although they had participated in the standardization of two of the notable mechanical achievements of the war, namely the Liberty engine and the motor truck, the tractor industry was not old enough to submit to the process of standardization. It appears that only about 5 to 7 per cent of the potential tractor market has yet been filled by tractors, and if that is true, we have not even 5 to 7 per cent of the knowledge that will be available in regard to tractors.

Time Not Ripe for Standardization

We are just starting on the era of great things for the tractor. It seems to me there never was a body of men who realized the value of standardization and its proper application as this Society does. I would deplore greatly our attempting to go too far at this time in standardizing the tractor, but I hope that no one will take my remarks to mean that we should not go as far as possible and as fast as possible in the standardization of numerous parts going to make up tractors.

PRESIDENT KETTERING:—Mr. Strite spoke about those in the automobile field going into the tractor business. There is one thing they have got to learn—that a tractor pulling three plows does not coast very much of the time.

Mr. Ford once said to me, "I didn't think there was so much difference between an automobile engine and a tractor engine." I do not believe he has found out all the differences yet.

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Keeping Dust Out of Carbureters

P. J. DASEY:—From my point of view, that of the engine business, the thing of importance to the internal-combustion engine industry and all those associated with it, is to devise some way of keeping sand and fine silicate dust out of the carbureters. Otherwise it will be of little use to standardize anything for tractors.

The second improvement needed is a positive governor that will control speeds within practical working limits. One idea today seems to be to install the gas engine regardless of the tractor, overload the tractor to the limit, run the speed up beyond what the tractor was originally built for, raise the engine speed beyond its extreme limit, and then blame the engine manufacturer for what happens. The engine should be sufficiently large to do the work without being run at the extreme limit.

By following these two suggestions the gas-engine tractor will be put on a much sounder basis. Otherwise I think we are going to see the red flag of revolt go up and steam becoming the prevailing power.

R. E. DAVIS (M. S. A. E.):—There is still a wide gulf between the automobile engineer and the tractor engineer. Each can learn something from the other. The automobile engineer has learned through years of practice to inclose the parts, and as far as possible to lubricate them automatically; in other words, to do away with the numerous grease cups which require attention, and make the tractor so that it will take care of itself.

The farmer's policy of lubrication is "let it squeak for oil first." We ought to provide a means of automatically taking care of lubrication. The farmer is in a rush in the busy season and he will neglect to tighten the grease cups. From some of the tractor designs I have seen I would not blame him if he did.

PRESIDENT KETTERING:—In studying this tractor situation it seems to me that we have, in addition to a great mechanical problem, one of the greatest psychological problems. There is little use in starting out to educate the farmer by printing an instruction book. We should make the tractor so that he will not have to study it out.

TRACTOR WEIGHT AND FUEL ECONOMY

ERNEST GOLDBERGER:—The law of averages, as a principle for finding the truth from catalogs of tractors on the market, fails completely in any problem of tractor engineering as in any branch of human activity that is

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still groping its way in the dark. In the early dawn of tractor design, farmers with insufficient engineering and manufacturing experience, and automobile engineers with only a hasty consideration of the farmers' needs, started to build tractors from opposite ends; hence the variety of sizes, types and weights. It is a mistake to suppose that a tractor, containing features obtained by averaging those of tractors already on the market, would present the progressive features for which there is such need at the present time.

To the question, "How many pounds pull per 100 lb. weight should a tractor develop at the drawbar?" almost any figure from 10 to 100 would be found to apply to the types on the market, and yet, with the exception of what the size of a tractor should be, this is the most important question for the manufacturer to know. That all these tractors have traction to a certain extent cannot be denied; therefore, the problem reduces itself to a question of economy. By reducing the weight of the tractor the fuel economy is necessarily increased, but on the other hand the cost of material, that is, the initial cost of the tractor is increased. What is, then, the most economical weight?

Suppose we consider three types of tractors: The heavy-wheel tractor which gets its traction mostly due to adhesion (friction) or lugs (not higher than 1¼ in.); the light-weight tractor, which gets its traction from the paddlewheel effect of the high spikes or grouters, which sink into the ground, hold the wheel for a while, and, in revolving, plow a shovelful of ground with every grouter; and, finally, the creeper type, which again gets traction mostly by adhesion, but sinks less into the ground.

German Motor-Plowing Contest

At one of the most scientific motor-plowing contests that I know of (Klein-Wanzleben in Germany in 1913), in which both European and American tractors of all types were well represented, every tractor was hauled over the field and the required pull and its weight compared. The official report (published in *Motorwagen*, July 20, 1914) may be considered reliable, as it was published before the war.

The results were:

1 Creeper tractor (American, one only) required a pull of 11.1 per cent of its weight for propulsion.

2 Heavy-wheel tractors required from 16.1 to 19.3 per cent.

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3 Light-weight tractors (all German make) required from 17.2 to 21.9 per cent.

Assuming as correct, averages of 11.1 per cent for creepers, 17.7 per cent for adherence wheel tractors, and 20 per cent for paddlewheel tractors, and assuming further a speed of 2.5 m.p.h., a tractor life of 4500 hours of work, a fuel consumption of 0.75 lb. of gasoline per hp-hr., and a cost of $0.20 per gal. for gasoline proportionate cost of oil included), we find that; if, by saving one pound of tractor weight *we do not increase the total cost* of the above three types of tractors (due to use of higher grade and lighter material) any more than 8, 12.8 and 14.5 cents respectively, then it is advisable to save as much weight as possible from an economical point of view.

The above takes into consideration the relation between weight of tractor and fuel consumption, which is the main, but not the only, item affected by the weight of a tractor.

The above figures are not of final reliability, but I give them in order to illustrate my proposed method of attack and to emphasize how important it is to have experiments conducted along this line and made public property.

Fewer Papers and More Discussion

L. J. Monahan:—The meeting brought out the fact that not over one or two papers at the most should be permitted for an afternoon, and more complete discussion should be encouraged. The trend of the papers because of practically no discussion was toward the four-wheel, two rear-drive wheel type.

Other types not referred to at all are running the four-wheel a hard pace in the market. The two-drive unit track-layer is gaining in popularity and the two-wheel universal type has developed a large market.

The business is not in shape as yet to consider standardization of one universal type. At the same time, a short discussion by a majority of the members present would aid in establishing a composite of opinions which in turn would show the drift the engineering aspect may be expected to take.

It would seem that a general engine type with standard mountings could be settled upon, as well as a final drive, also a standard transmission mounting applicable to the four-wheel, two-drive wheel, and two-drive track-laying type. It is beginning to look as though the track-laying type could be easily standardized in the simple two-track-drive without any separate front support. The

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four-wheel type can easily be standardized and the two-wheel type shows a development which would not appear to require radical changes in the future; hence there is no reason why it could not be standardized as a type. The vertical multiple-cylinder or opposed double-cylinder engine for this type, however, is unsettled.

FUNDAMENTALS OF TRACTOR ENGINE DESIGN

By H C BUFFINGTON*

The subject of tractor engine design seems simple at first thought, and immediately the general outline of a four or six-cylinder engine flashes through one's mind. We picture a smooth iron-clad masterpiece solidly mounted upon the frame of a "racy" looking tractor. We do not at first consider the carbureter, magneto and other contrivances necessary to make it run.

Presently we ask ourselves, What are the essentials of a successful tractor engine; what have we gathered from actual experience, and not from theory? What I have in mind is not an engine of high speed and light construction, nor is it a heavy one or two-cylinder engine; it is a four-cylinder, with certain elements of construction which from experience have proved their worth. These elements of vital importance are the ones upon which I shall dwell.

Crankshaft—This is the essential element around which the engine is constructed and therefore deserves first consideration. Too often this part has been neglected in the fight for lightness and cheapness. Personally I prefer the three-bearing shaft. There are, however, some points in favor of the two-bearing type, in that the two center cylinders can be placed more closely together. Doing away with one bearing is an advantage, but on the other hand a heavy shaft is necessary and generally speaking the three-bearing type is best suited for large engines.

The crankshaft stress should not exceed 12,000 lb. per sq. in. for a 300-lb. per sq. in. explosion pressure. This will give a good solid shaft free from the vibration so destructive to bearings.

Crankpin Bearings—A good proportion for the crank-

*Motor Engineer, Minneapolis Steel & Machinery Co.

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pin bearings is to take 57 per cent of the cylinder diameter for the pin diameter and 1.2 of the resulting diameter for the length. This will give a projected bearing area of half the cylinder area, thus keeping the pressure down to 600 lb. per sq. in., using the 300-lb. explosion pressure. The main-bearing stress, starting from the front end, should not exceed 400 lb. per sq. in., 400 lb. for the center and 250 lb. for the rear, taking the centers of the bearings as the points of reaction. The rear bearing pressure is low, but this is necessary to take care of the extra pressure caused by the transmission shaft if it happens to be out of line with the engine, which is often the case.

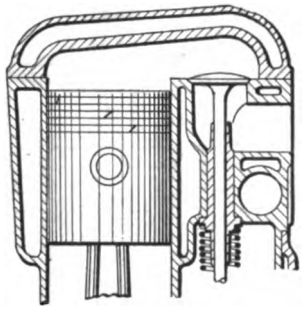

FIG. 1 — L-HEAD CYLINDER WITH HEAD PARTING PLANE ABOVE PISTON FACE

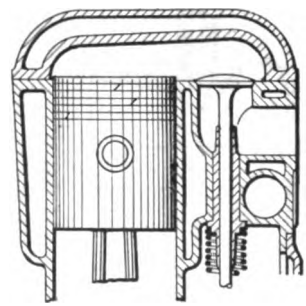

FIG. 2 — L-HEAD CYLINDER WITH HEAD PARTING PLANE SLIGHTLY BELOW PISTON FACE

Removable Head—Whether the cylinders should be of the L-head or valve-in-the-head type is a matter of choice. Those who prefer the L-head perhaps use it because, as I heard it expressed, "it is a good old family horse." On the other hand, the valve-in-the-head is chosen by the more particular, who do not want to be outdone on points of efficiency. The cylinder head should part in line with, or a little below, the top of the piston and not above, as the latter method makes it hard to clean; and the piston cannot be removed without first removing the carbon from the cylinder walls. There are several important reasons why the removable head should be classed as essential; for example, the possibility of cleaning carbon, grinding valves, doing away with the troublesome valve-cover plug or valve cage, and making it possible to adjust the compression for high altitude. Figs. 1, 2, 3 and 4 show this difference and also illustrate how the adjustments for different compressions can be taken care of much more easily. Many tractors are working in altitudes as high as 8000 ft. and the changing

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of the head is a simple matter as compared with the changing of pistons.

Water Jackets—The water space around the head should be liberal, as, unlike the automobile engine, a maximum load must be expected for a period of hours. An even thickness of combustion-chamber walls is essential for the same reason, and means for checking up the thickness should be seriously considered. If, for instance, there be a hot spot in one cylinder, the carbureter must be adjusted for that, or the result will be disastrous, causing either preignition in one cylinder or excessive fuel leakage past the other three pistons.

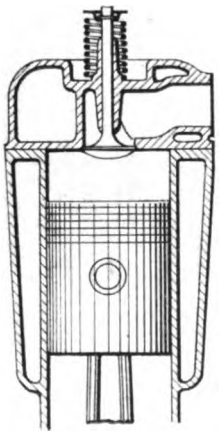

FIG. 3—VALVE-IN-HEAD CYLINDER WITH HEAD PARTING PLANE ABOVE PISTON FACE

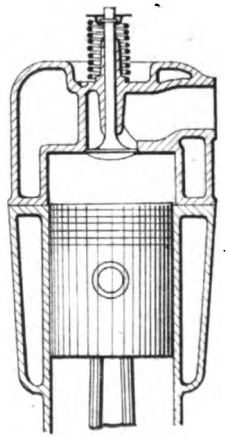

FIG. 4—VALVE-IN-HEAD CYLINDER, HEAD-PARTING PLANE SLIGHTLY BELOW PISTON FACE

Long Stroke Preferable—Theoretically, a tractor engine should not be pulled down below normal speed, but this is often done; in fact, at least 50 per cent of the time the operator will have his engine overloaded. For this reason the stroke should be at least 1.4 times the cylinder bore. The long-stroke engine is also desirable, because the difference between the engine speed and that of the final drive is not so great, and it is more convenient in minimizing the transmission-gear diameters.

Flywheel—The transmission itself has a great deal to do with determining the flywheel capacity, and the higher the efficiency of the transmission, the greater should be the increase in the flywheel energy. The less efficient transmission acts as a shock absorber and decreases the shock on the crankpins, while on the other hand the shock is transmitted almost directly to the crankshaft.

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It can then readily be seen why some engines with small flywheels do not have half a chance to stand up. In calculating the flywheel dimensions, double the horsepower should be taken with an allowance of 10 per cent variation in speed.

Speed Governing—The control on the engine should be such as to limit the speed to about 10 per cent above normal.

Engine Rating for Kerosene—This should be based on 800 ft. per min. piston speed, and not be over 75 per

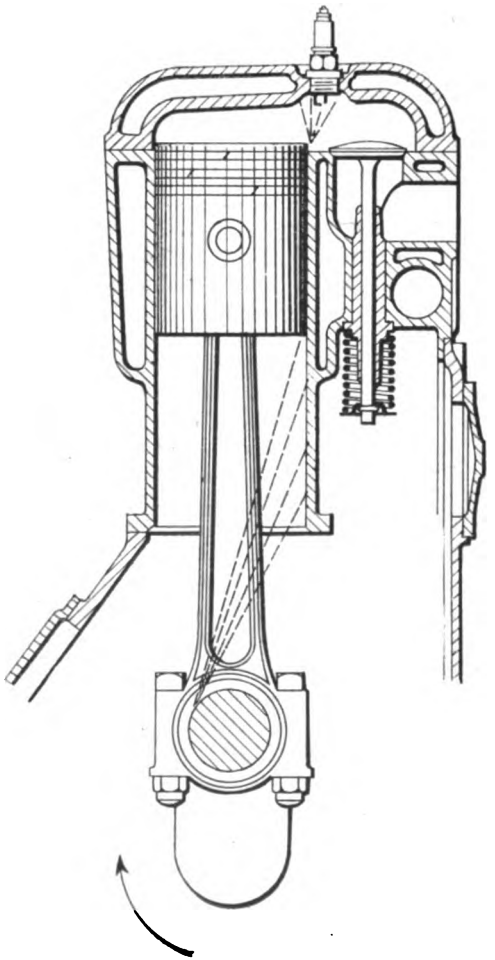

FIG. 5—UNSATISFACTORY LOCATION OF SPARK-PLUG WHERE IT IS SUBJECT TO IMPINGEMENT OF LUBRICANT

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cent of the engine's maximum capacity with gasoline. Laboratory tests have shown that the power output with kerosene is within 8 per cent of that with gasoline, but this cannot be expected under actual working conditions.

Fuel Economy—When pulling its rated load before leaving the test block a tractor engine should not use more than one pint of kerosene per horsepower-hour. This fuel should have a gravity not lower than 42 deg. Baumé, and an open flashpoint of not over 120 deg. fahr.

Engine builders today are at a disadvantage in not having a standard fuel with which to develop their product. I suggest that the Society establish a standard kerosene and a standard gasoline.

Manifold Heat-Control Essential—One reason why it is difficult to run tractors successfully on kerosene in the country is the grade of kerosene used in small towns. This kerosene might be termed lamp oil, and although the specific gravity is not so low, the open flashpoint is very high. In order to vaporize this fuel for use in engines, it must be heated to a very high temperature, which, of course, reduces the power output. This fuel gives a very good light, and there is no danger of it exploding, but it costs more than the so-called distillate, which in most cases is an excellent fuel.

This point pertaining to difference in fuels is mentioned to emphasize the necessity of controlling the heat between the carbureter and cylinders.

When it is necessary to use water with the fuel, the control of the former should be automatic, because it is our experience that the farmer has had considerable trouble during the first few weeks' use of engines not having automatic control of water.

Spark-Plug Location—I have often been instructed as to the proper location of the spark-plug, but after many experiments and sad experiences, have proved that the trouble in securing such a location is due largely to lubrication.

Without doubt 50 per cent of magneto troubles will be avoided by placing the spark-plug properly, and yet engines are designed without attempting to determine this. Without hesitation the designer will place it over the intake valve on an L-head engine or on the intake side of the valve-in-the-head type. Spark-plugs should not be placed near the line of the cylinder bore, where oil from the piston will splash on them (see Fig. 5), or in line with the intake gases when using kerosene as fuel. A careless operator will use either too much

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fuel or too much water with the fuel, and improper ignition will result when the plug is directly in the path of the gas flow.

The direction of the crankshaft rotation will sometimes make it necessary to shift the plug to the opposite side, because the excessive oil (Fig. 5) thrown from the crankshaft will always lodge on one side of the cylinder. If the plug is on this side or in line with this side, it will become fouled much more quickly. The location over or near the exhaust valve has often been criticized, but aside from the fact that the spark-plug itself will have to be replaced more often, this is the most reliable position.

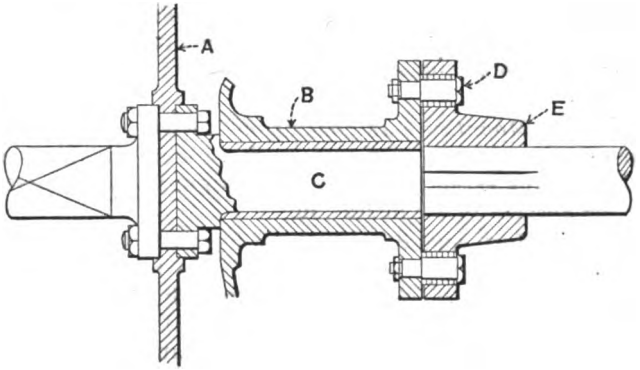

FIG. 6—FLEXIBLE CONNECTION BETWEEN ENGINE AND TRANSMISSION
A—Flywheel, B—Clutch-Driving Spider, C—Stub Shaft Extension, D—Flexible Driving Pin, E—Flange of Transmission Shaft

Lubrication—The tractor engine furnishes the hardest lubrication problem in engine design at present, and will continue to do so until a better method of using kerosene as fuel is employed. An excellent oiling system has been developed for the automobile, but it has turned out to be a miserable failure when used with kerosene under actual working conditions in the field.

We dislike to throw away lubricating oil, and every effort should be made to prevent such loss. It is cheaper, however, to use the lubricant once only, and then let it drain from the crankcase, than to cause the deterioration of the engine by too great economy of oil. Time is lost in making adjustments and repairs caused by the used-over oil, as it gradually becomes thinned by the fuel passing the pistons. The most reliable method of lubrication is to force fresh oil to each moving part, crankpin

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bearing, main and camshaft bearings, gears and cylinders, and then drain it into a reservoir from which it is drawn from time to time.

Every care should be taken to clean the air drawn through the carbureter and breather. The oil supplied the crankpin must pass through a drilled hole, so that any dust entering the crankcase will gradually pack and clog the passages. Any method, whether force or splash, in which oil is used twice, will cause rapid wear. This is not the case with carefully operated engines, but we cannot expect unusual care from the average operator.

Accessibility—Handholes placed in the side of the crankcase, so that the bearings can be inspected, are quite necessary and the base-pan should be light and easily removable. Every provision should be made to have wearing parts so placed as to be easily removed and to avoid crowding small nuts into corners and out-of-the-way places.

Engine Mounting—In conclusion, I wish to say a word about mounting to the tractor designer; this may not relate to the essentials of engine design, but is necessary to the success of its operation.

Every effort should be made to have the engine accessible, not only on the carbureter side but all around, especially to allow clearance underneath. Allow plenty of room for removing the base-pan; if possible avoid fastening tractor parts to cap screws and nuts used in fastening engine parts, especially to the cylinder-head bolts. This only tends to discourage the operator in making necessary adjustments.

I cannot lay too much stress upon the importance of having a flexibfle connection between the engine and the transmission drive-shaft. It has been necessary to provide extra large bearings at the flywheel and to take care of the vibration caused by the transmission and engine shaft being out of line, but this destructive feature can be remedied. I suggest that a self-contained clutch be mounted upon a stub shaft bolted to the flywheel flange bolts, and also that a flexible coupling be placed between the clutch and the transmission. This construction, Fig. 6, not only takes the extra pressure off the crankshaft, but also allows the transmission bearing to run freely.

THE DISCUSSION

R. E. DAVIS (M. S. A. E.):—The old type of heavy tractor has the fault of being open and having too many places

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that require attention. I refer to the open valves which are subject to showers of dust under the average field working conditions. This shortens their life greatly, far more than one might expect. On the other hand, the average automobile type of tractor engine, as we might call it, is inclined to be too "skimpy" in the bearings, and as the paper just read has brought out, sufficient attention is not given to lubrication. If we could combine the idea of large bearings and inclosed construction to keep out dirt, with maintaining a generous film of oil, the life of the tractor would be prolonged anywhere from three to five years; this applies especially to the valves, which are vital parts of the tractor and is particularly true of the horizontal valve which tends to rub continually against one side of the stem guide and chafe the seat at every stroke.

Fuel Problem

THOMAS J. LITLE:—I believe that the biggest problem today in tractor engine design is the fuel problem. We are trying to use kerosene in gasoline engines and are not succeeding very well, because we hear of the dilution of crankcase oil and consequent destruction of the bearings. When we try to use kerosene in the old-type gasoline engine, and send fuel into the intake passages in the form of a rainstorm, a great deal of it drops out, much of all that gets into the cylinders never burns but merely wets the cylinder walls and is forced down by the rings. The real fuel problem is to gasify or vaporize the fuel before using it. Not only kerosene, but a great many other fuels are going to be used after this war. I have been spending all my time for the last two months investigating this very problem. Alcohol and benzol and mixtures of the two, and a good many other engine fuels will be employed, but the engine designer must not leave it all to the carbureter manufacturer. He must bear this in mind and avoid abrupt turns in the intake passages because the fuel cannot be sent around too many corners.

J. C. SLONNEGER (M. S. A. E.):—The company I am with has not yet attempted to solve the kerosene problem, and I am not in position to speak about it. A really successful tractor engine has not been built to burn even gasoline, and we are merely adding to our troubles in attempting to burn kerosene. I am sure that if the manufacturers of tractors would try to solve the problem of burning gasoline satisfactorily, and go after it with as much energy as they have after kerosene, we would have a much better tractor engine today.

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Encouraging Alcohol Manufacture

G. L. Lewis (M. S. A. E.):—The war is bringing us face to face with a problem that might have been held off for another generation, but the enormous demand for fuel oils has increased by leaps and bounds, until it is reported there will be a shortage of over 1,300,000 bbl. this year. It is also reported that the Standard Oil Company took out of its reserve over 830,000 gal. of gasoline last year.

In the development of our internal-combustion engine the minutest details have been gone into microscopically, so to speak, and with the greatest amount of criticism, that we might make an engine as nearly perfect as possible, yet we have considered the fuel supply only indifferently, or only as it was forced upon us. We have scarcely concerned ourselves about the source of supply or permanency of the supply. When some far-seeing mind attempted to bring up legislation whereby a substitute could be manufactured, a strong lobby immediately set to work to defeat the measure.

The time is opportune and ripe for the Society of Automotive Engineers to turn its attention to alcohol legislation. I think it is our patriotic duty to assist in forcing Congress to pay heed to the needs of the people. Now is the time to enact laws authorizing the manufacture of alcohols so that relief from the high cost of fuel can be had in the home as well as in the engine industry.

The world is in a critical condition, and we should develop our artificial as well as our natural resources, for which we have an unlimited supply of raw materials. There is scarcely a house in the land that does not throw away enough raw material to heat, light and cook with, if proper legislation were enacted. Convert our garbage plants into alcohol stills, and the utilization of this now waste material would be worth millions to our Nation.

Oliver E. Barthel (M. S. A. E.):—There is apparently a lack of understanding of the fundamental principles underlying the proper design of a tractor engine. This was evident at the Fremont demonstrations last summer in the absence of a single example of good tractor engine design in any tractor exhibited.

Automobile Engines Unsuitable for Tractors

There were some tractors at Fremont equipped with engines that were excellent examples of good automobile and truck engine design, but, strange as it may seem, the engine builders and tractor makers are deceiving them-

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selves into believing that they can "get by" with the coaster type engine, designed to operate under full load less than 25 per cent of the time.

Engineers and designers accustomed to designing automobile and truck engines along lines regarded as standard practice seem to lose sight of the fact that so-called standard practice is the result of the empirical or trial method, and for economic reasons has been considerably modified from time to time to reduce weight and to meet economical quantity production methods. It is true that trial and experience have determined approximately both rational and empirical formulas for the most essential parts of automobile and truck engines and from existing envine constants have been derived and more or less conveniently arranged for designers' use. It, however, does not follow that formulas which have proved satisfactory for the type of engine required to operate under full load only a small part of the time should be followed in designing an engine to operate under full load the greater part of the time.

In designing an engine for a different service, such as for a tractor, it is necessary to go back to first principles in machine design and develop formulas, rather than assume that existing formulas on automobile or truck engine design should apply equally well to tractor engine design.

Reserve Power

I venture to say that the selection of the ultimate type of tractor engine will not be determined by its prevalence on automobiles or, by any stretch of the imagination, that the tractor should be camouflaged to suggest the one non-essential attribute of a tractor, namely racing speed. As to number of cylinders, the determining factor is the minimum piston area permissible per cylinder at normal speed for the drawbar pull required. In other words, the cylinders must be of sufficient diameter to provide for the sharp momentary overloads that the tractor will be subjected to from time to time while operating under full-load conditions at normal speed. Reduced to a practical basis, it means that a tractor whose drawbar pull requires an engine of about 26 b.hp. or under must necessarily be of the two-cylinder type in order to have sufficient reserve energy to overcome a suddenly-applied overload, while tractors requiring engines of over 26 b.hp. are best served by the four-cylinder type.

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Two-Cylinder Engines

H. L. Horning in his paper on "The Ultimate Type of Tractor Engine"* enumerates the difficulties inherent in the design of two-cylinder opposed engines as follows:

1—The tendency of the shaft to revolve longitudinally in the plane of the crankpins, due to centrifugal, inertia and gas forces. This tendency loosens the main bearings by throwing the shaft out of line. The crankpin bearings ride alternately on one side of the pin and then the other, causing damage and necessitating frequent adjustments and replacements.

2—Great weight of reciprocating parts compared witn the brake horsepower, augmenting the difficulties enu merated under (1).

3—The high centrifugal forces as compared with the brake- horsepower, resulting in effects as listed under (1).

4—Difficulty in cooling the large piston and valve heads necessary in obtaining 20 hp. per cylinder.

5—Difficulty of maintaining a perfect mixture in the long intake header as well as the impossibility of attaining high volumetric efficiency with the high velocity necessary for maintenance of mixture quality.

6—Difficulty in lubricating all parts sufficiently without over-lubricating the cylinder.

7—The disadvantageous location of the valves.

8—Infrequency of impulse, the lack of balance and its influence on the transmission mechanism.

The difficulties listed under numbers 2, 3, 4, 6, 7 and 8 need not exist in a properly designed two-cylinder opposed engine, while numbers 1 and 5 are only partially true and can be sufficiently overcome so that the final power output and efficiency will compare favorably with other types of engines.

Four-Cylinder Engines

In 1897 I worked on the design of a four-cylinder valve-in-the-head marine engine similar to the one shown in Fig. 4 (with head-parting plane slightly below piston face), and after observing a number of these in operation, designed the next engine with the head as shown in Fig. 3 (head-parting plane coinciding with inside face of head) for the reason that it provided machined combustion chamber walls without additional operations, insured a uniform compression space in all cylinders, overcame the liability of the piston striking the gasket or

*Transactions S. A. E., 1917, Part 1, page 65

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head without the necessity of providing an excessive counterbore, and still possessed the advantages claimed for the design of Fig. 4.

It is a waste of time to even consider the two-bearing type of crankshaft in connection with four-cylinder tractor engines, for the reason that many years ago when the first two-bearing crankshaft four-cylinder automobile engine was designed, which by the way had the cylinder cast in block, it was found that this type of crankshaft could not be used successfully where the distance between main bearings was over 19 in. Further, where there are several throws between supporting bearings, as in the two-bearing, four-throw crankshaft the crankpins must necessarily be made larger in diameter than would otherwise be necessary to perform their function as a bearing, in order to withstand the bending stresses. This produces a greater rubbing speed and causes undue heating of crankpin bearing.

Lubrication Difficulties Caused by Dust

The conditions under which the tractor engine operates are so radically different from those of the automobile engine, that it is pathetic to attempt to apply the automobile oiling system to the tractor engine without first analyzing the conditions to be dealt with. Granting that a method of using kerosene will be developed that will eliminate all chance of raw fuel getting into the crankcase, there still remains the dust problem which the air-washer does not entirely solve even if fitted to the crankcase breather pipe. It is only a question as to the relative density of the dust cloud in which the tractor is moving as to how soon the engine will begin to inhale mud through the washer. Dust and mud mixed with oil have the same abrasive action and will seriously damage the engine, if there is not a constant washing action of the oil to keep the abrasive material moving away from the wearing surfaces. This can be accomplished by forcing fresh oil to the parts that require lubrication and letting it drain from the crankcase into a container that can be readily removed.

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TRACTOR POWER TRANSMISSION

By E R GREER*

When we speak of a tractor transmission, we generally mean the entire power-transmitting device from the engine to the ground, including the pulley for belt work. Tractor transmissions are still in an early stage of development. They are composed of a number of groups of parts that are arranged differently in nearly every case, so that they cannot be classified without dividing them or making a special class for nearly every tractor now being manufactured.

There are differences of opinion among tractor engineers regarding many of the most vitally important features of tractor transmission. Some believe in rough open gearing and large plain bearings, as opposed to those who insist on cut hardened inclosed gears and anti-friction bearings.

The question of wheels for traction vs. track-layers is far from settled. The right size of wheels for wheel tractors seems further from being agreed upon than anything else, and it is one of the most fundamentally important things in tractor design. The design of a tractor transmission is affected greatly by the design of the wheels or track-layers that are to give the traction. The matter of traction is of first importance to the successful tractor, and it is here that opinions differ widely.

The track-layer with its large surface contact has an advantage on marsh land or sand, where the top soil is tougher than the ground beneath, but is at a disadvantage when, just after a rain, the ground is muddy on top and harder below. It can bridge a small ditch but becomes blocked by one too large to bridge, and side hills or side draft cannot be taken care of as efficiently as when wheels are used.

Wheels, simple and not expensive, must be of sufficient size and provided with lugs to give good traction. If a tractor wheel once starts to dig it will not climb out of its hole unless the load is released. On smooth hard ground a small wheel gives efficient traction, but on newly plowed sandy soil or on land where there are washed-out ditches a large wheel must be used. The matter of lugs is of utmost importance. On some kinds of ground a

*Assistant Superintendent, Emerson-Brantingham Company

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spiral cleat at about a 45-deg. angle will give double the tractive effort that can be obtained from the same wheels with straight cleats or spikes.

For efficient general farm work with wheel tractors of from 8 to 15 drawbar hp., the drive wheels must be 5 ft. or more in diameter, and the problem of the best way to drive the wheel is important. The effort being applied to the ground, it follows that the rim is the natural place to apply the power, but the rim runs in the soil, so that it is necessary to apply it at some point far enough from the rim to be protected from the dirt. If the drive is carried to the axle the torsional strains become severe, and good construction is expensive and difficult. From all indications the inclosed internal-gear drive is likely to become the standard traction-wheel drive.

The next important item is the location of the drive-pinion. The one best location is a point forward and a little below the wheel axle. When the pinion is in this location the driving force is carried directly to the ground, so that the load on the drive-wheel axle bearings due to the power delivered is opposite the load due to weight, and the amount of load is the difference between the driving effort delivered by the drive pinions and the weight of the tractor that is supported by the axle when the tractor is standing. In other words, the drive-axle bearings are practically floating when the tractor is pulling a certain load. At less load the bearing pressure is downward and at greater load the pressure is upward. The strain of the drawbar pull is carried by the drive-pinion bearings through the pinion teeth to the drive-wheel rim and the ground. It is the lifting effect on the front end of the tractor that results in its forward movement. A number of tractors have enough power to raise the front end clear of the ground so that the front-end weight is the limit of their pulling power. These tractors are difficult to guide and are of no use at all on hills. Enough dead weight must be carried on the steering wheels so that the tractor can be guided on the steepest hill.

The front-wheel drive offers an easy solution for hilly ground as in this case the reaction from the load is downward on the rear steering wheels. There is no tendency to tip up, and the harder the pull the more positive is the control. The rear wheels must be of ample size to carry the additional load, and the difficulty is in arriving at an arrangement to handle the implements when the front wheels are the drivers. While the steering mechanism

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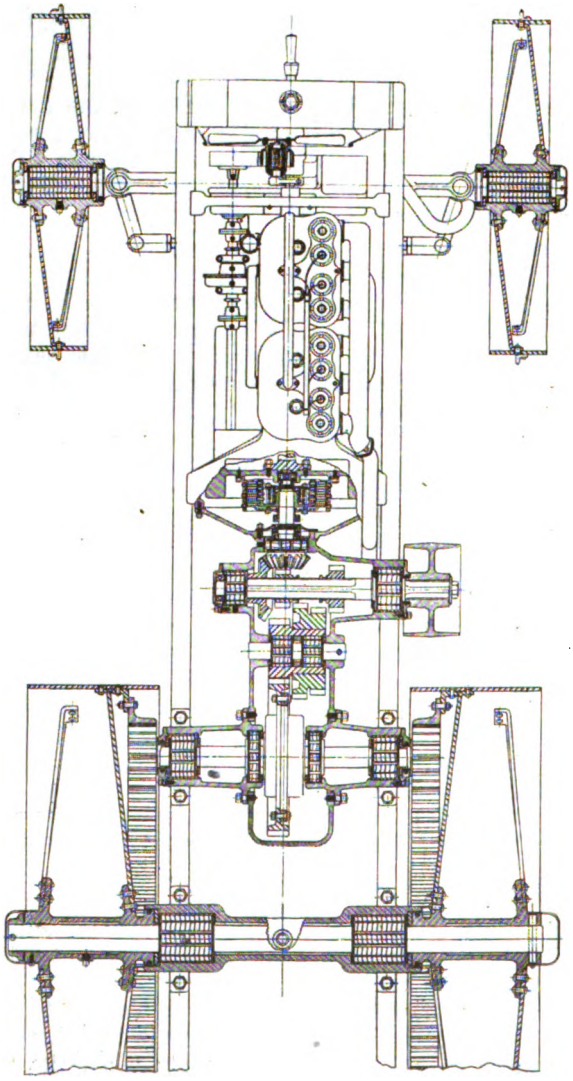
may be complicated, the power transmission can be simple, the weight can be placed mostly on the drive-wheels, and ditches and hills can be easily negotiated. The four-wheel drive immediately suggests itself, but the mechanical complication and expensive construction make it prohibitive.



HYATT ROLLER BEARING CO. TRACTOR BEARING LAYOUT

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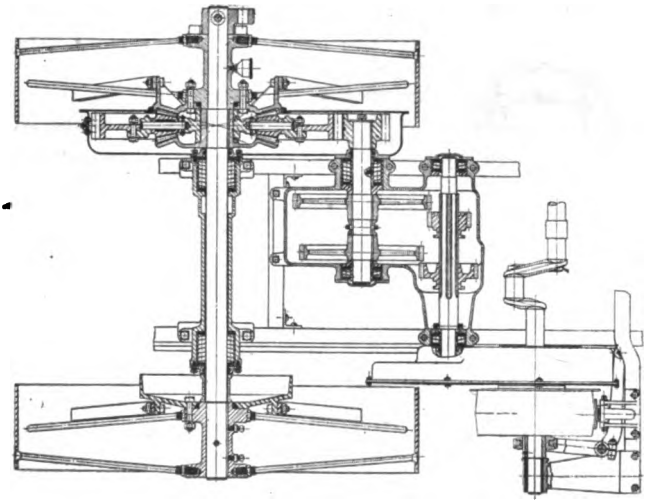
The protection of the working parts is not easily accomplished in tractor transmissions. An argument used by those who favor open rough-cast gearing is that the farmers cannot keep transmission grease or oil clean and that some grit is always put into the case at every filling. They make the claim that as long as parts cannot be kept clean they wear out and may as well be made simple and open, to operate without lubrication or when covered with dirt.

One unprotected drive, known as the roller pinion



TRANSMISSION AND TOP DRIVE OF CASE 9-18 TRACTOR

drive, has been extremely successful on tractors. It can be made to last a season, is cheap, and is easily replaced. Perhaps the worm-drive gear can be taken as the opposite extreme because the excessive tooth pressures, together with the vibration and dirt, prevent its being a success, no matter how carefully it is built.

As the users of tractors become more skillful and cease to abuse their machines, thoroughly protected well-made parts will become more general. Today simplicity is essential if a tractor is to be successful.

Anti-Friction Bearings

A notable advance in tractor transmission design is in anti-friction bearings, which are becoming universally used. It has been found that an ample, non-adjustable,

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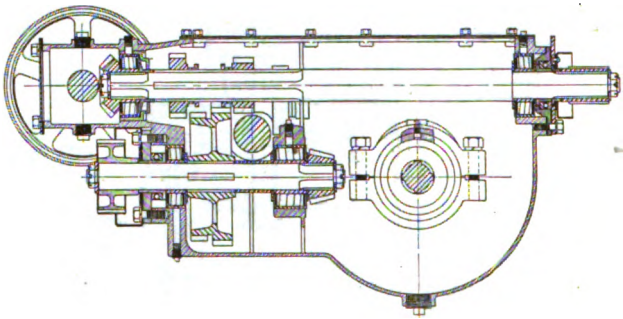
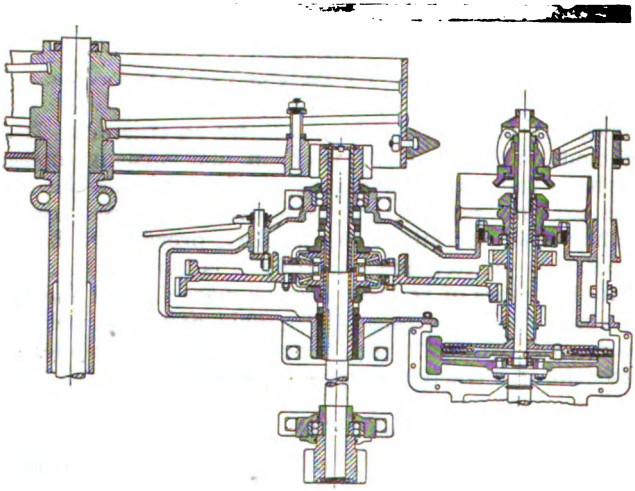
anti-friction bearing can last indefinitely in tractor service. Transmission-bearing trouble has been a common tractor fault in the past, and its elimination is a long step ahead.

The shafting, as a rule, is made of ordinary mild steel, because gears and bearings generally have to be of such size that shafts are amply large when this material is used. It is important that splines be used instead of keys.

The standard type of automobile or truck transmission does not meet tractor requirements at all. The gear reduction is too small, no belt pulley is provided for, the



VERTICAL SECTION THROUGH EMERSON-BRANTINGHAM TRANSMISSION



FARRETT CLUTCH TRANSMISSION AND DRIVE

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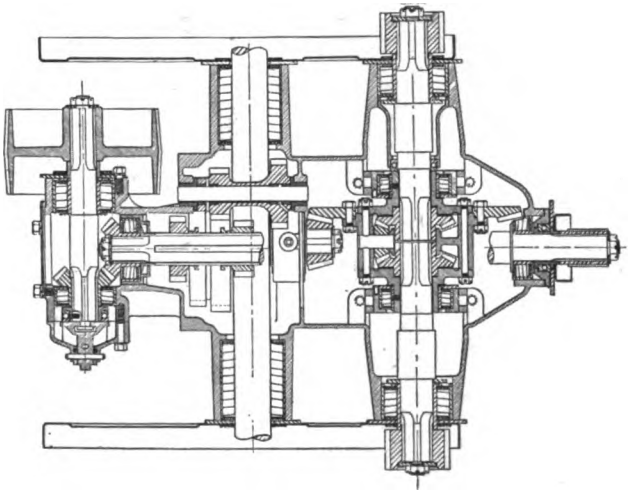
change-speed arrangement is wrong, the parts are not large enough, and the type cannot be applied to a good tractor design.

The total gear reduction in a tractor varies between 30 to 1 and 100 to 1, so that several reductions have to be used, although in some cases entire transmissions are built as a single unit. It is best to discuss each reduction separately, and to consider especially the final reduction by itself.



HORIZONTAL SECTION THROUGH EMERSON-BRANTINGHAM 12-20 TRANSMISSION

In general all tractor transmissions are driven from the engine through a friction clutch. No universal joints are used, although clutches are designed to take care of any misalignment. Cone, disk, shoe, band clutches, and friction drives are in use. The cone clutch and the internal expanding shoe clutch are most common.

First Reduction Gearing

First-reduction gearing may be of almost any kind. The reduction varies from 1 to 1 to as much as 8 to 1. The change in speeds and the reverse are often worked in on the first reduction, but it is also common to use the second reduction. When the engine shaft is not parallel with the drive-wheel axis the first reduction gears are

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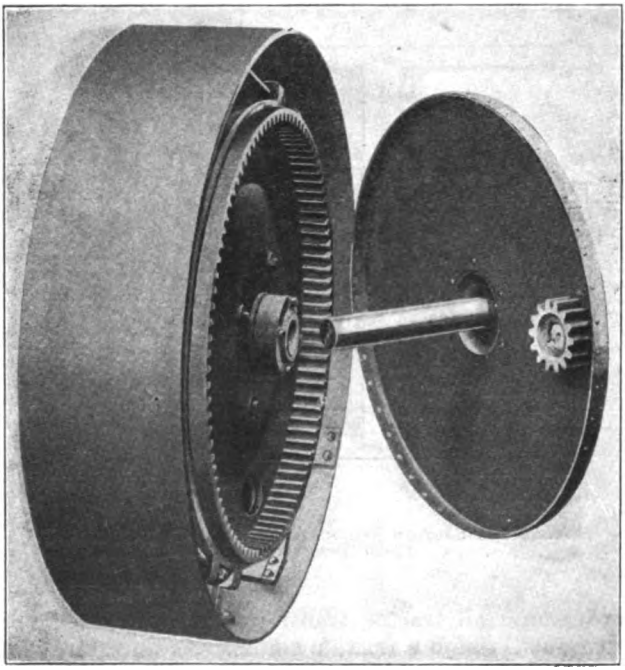
often of the bevel type. There is considerable advantage in having the bevel gears the highest speed gears. The cost is reduced and a chance is provided to get a belt pulley on a fairly high-speed shaft without the necessity of using a separate set of gears and a clutch.

When spur gears are used for the first reduction, the change-speed gears and reverse are generally a part of it as they can be much smaller and cheaper in this position than in the next reduction. This arrangement is



MINNEAPOLIS STEEL AND MACHINERY CO. INTERNAL GEAR FINAL DRIVE

general on tractors in which the engine is set crosswise; it is also used to some extent when the engine shafts are set lengthwise of the tractor, even though this arrangement necessitates a larger and more expensive set of bevel gears in the second reduction and also the use of a separate set of gears for the belt pulley.

Change-speed gears are usually arranged so that the pinions slide on the drive-shaft and engage large gears

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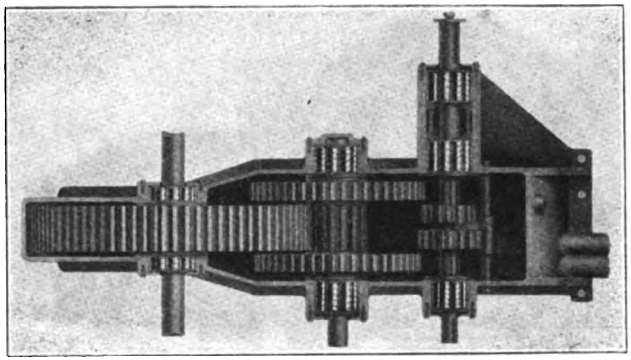
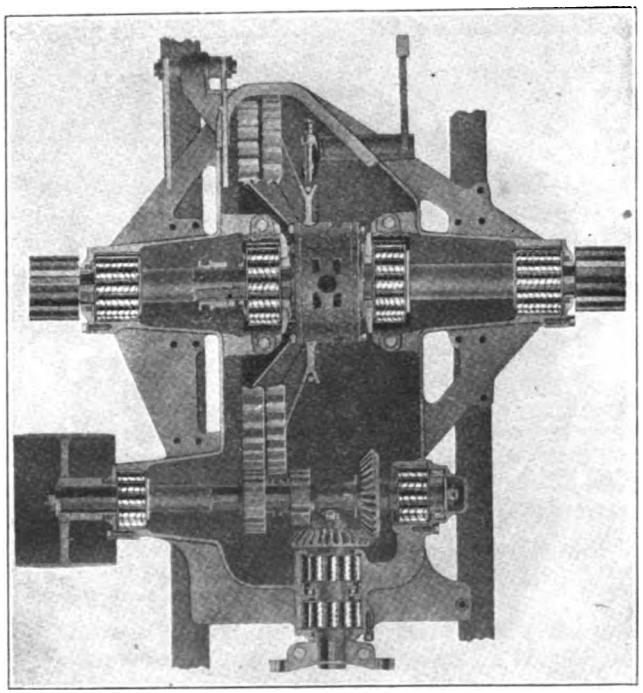
on the driven shaft. No divided shafts or countershafts, such as are found in automobile transmissions, are used except for reverse gears. Thus all gears are disengaged,



TRANSMISSION OF THE GRAY TRACTOR



"BIG FOUR" 20-35 TRACTOR TRANSMISSION

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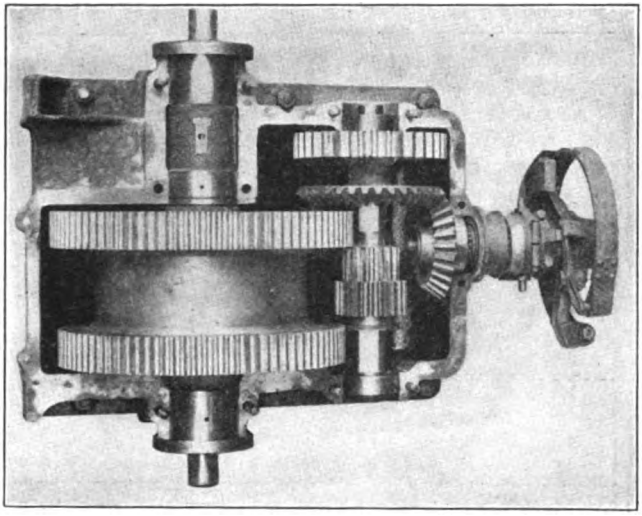
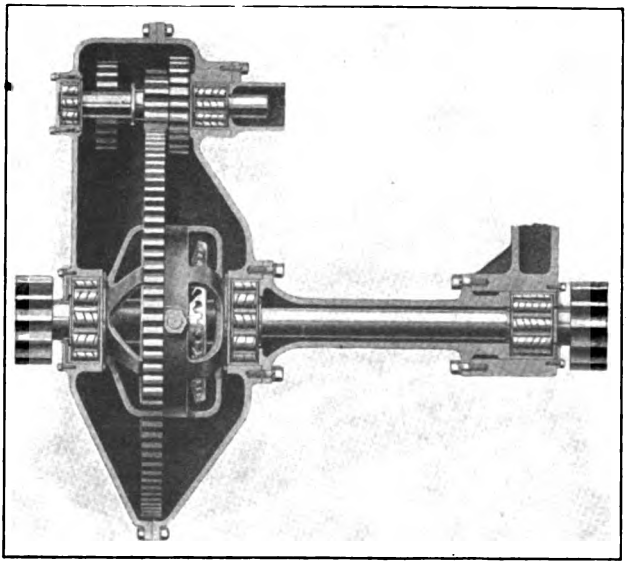
except the ones actually in use, and no idle gears or extra reductions are needed for different speeds.



MINNEAPOLIS STEEL & MACHINERY CO. CLUTCH AND TRANSMISSION



MOLINE-UNIVERSAL TRANSMISSION

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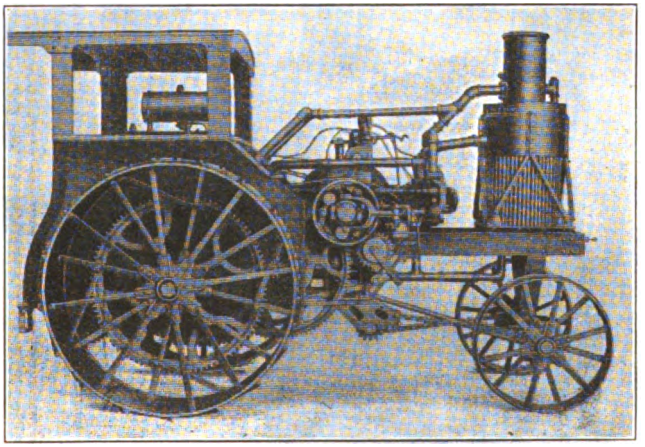
Another advantage in this arrangement is that all change-speed gears are disengaged when the belt pulley is to be used. Tractors are used for belt work on long continuous runs and gears or shafting running idle are undesirable because they waste power and require special lubrication.

Second Reduction Gearing

The second reduction is sometimes the final drive, although this is not often the case. Usually it is the drive to the shaft that carries the differential. If the differential can be placed in this position ahead of the last reduction considerable expense will be saved because all the parts can be made much smaller.



AVERY TRACTOR, SHOWING LARGE EXPOSED TRANSMISSION GEARS

Differentials have to be made strong and of ample size, because the traction members are always slipping, so that the differential parts are working all the time. Differential locks are considered good practice. Brakes for holding one side for short turning are often used, but brakes for stopping, while necessary, are not important.

Third Reduction Gearing

The third is usually the final reduction; it is the most important part of a tractor transmission. The load and vibration on the teeth and bearings generally cause wear of these parts first, and as they are large, expensive, and

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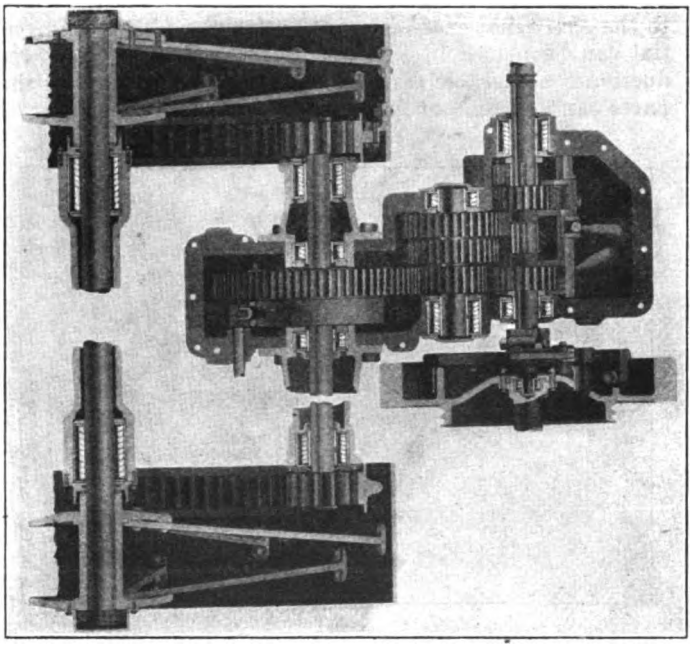
difficult to replace it is necessary they be made to give the best of service.

When there are more than three gear reductions, the extra one is used either to gain compactness or to provide for some special arrangement of parts.

Tractors are used for plowing a large part of the time and generally are operated at speeds of from 2 to 2¾ m.p.h. Many have only a single speed, but most of the late designs have two or three speeds. When there are



TRANSMISSION AND INTERNAL GEAR DRIVE OF THE WATERLOO BOY TRACTOR

three speeds one is lower and one higher than the plowing speed. When two speeds only are provided the second is usually slower than that for plowing. The range between speeds is nearly always between 30 and 40 per cent.

A tractor transmission is really a series of compromises, each part being affected by all the others; hence it is not surprising that there are such differences in design. A different location of the belt pulley alone will materially affect the entire transmission arrangement. At present there is little chance of formulating valuable

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transmission standards. It will require more time and many tests to prove what is best.

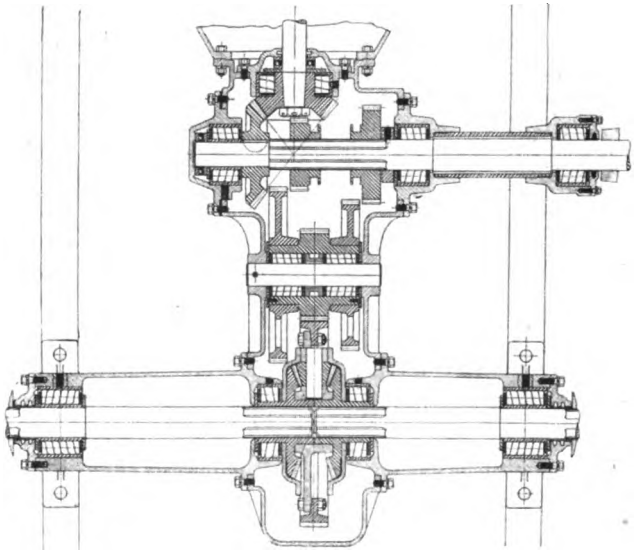
THE DISCUSSION

CHARLES A. TRASK (M. S. A. E.):—Thousands of tractors with friction transmissions are in successful operation, and I feel that this type is entitled to more than passing mention, in view of the complete information given by Mr. Greer covering various forms of gear transmissions.

MR. GREER:—I was not able to obtain any illustrations showing tractors with friction transmissions, which is the reason they are not covered in my paper.

ANTI-FRICTION BEARINGS FOR TRACTORS

R. E. DAVIS (M. S. A. E.):—Most of the illustrations in Mr. Greer's paper show anti-friction bearings. This



R. D. NUTTALL CO. TRANSMISSION

shows the trend of thought to be in favor of fuel economy, which is very fitting at this critical time in our Nation's history. The question of economy on drawbar work is of great importance, not only to the user but to the entire

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world. The amount of power loss in a plain bearing is from 10 to 20 times greater than the loss in an anti-friction bearing, according to available information and tests.

Taking the smaller figure, and giving the plain bearing the benefit of the most favorable conditions, the difference in power loss is still great enough to warrant our careful attention. It is great enough on a tractor of 20 b.hp. to increase the drawbar capacity and efficiency 40 to 60 per cent over the capacity and efficiency with plain bearings, all other things being equal.

At once the question arises: "If the engine has power enough to slip the wheels with plain bearings what is gained in capacity by the added saving in power?"

Nothing in many cases, but this merely proves that the design is not properly balanced to take advantage of the engine's power when anti-friction bearings are substituted. This condition could easily be adjusted and advantage taken of the increased drawbar power of 40 to 60 per cent, still reserving enough to slip the wheels.

Where the old rating for such a tractor was 10-20, with plain bearings in the transmission and axle, the rating could be 15-20 for anti-friction bearings. Of still greater importance is the question of drawbar economy. Taking again the 20 b.hp. tractor previously referred to, the difference in power loss between plain and anti-friction bearings amounts to 5.38 hp. at full load. The plain bearing losses are distributed as follows, based on 10-20 rating:

	Drawbar Per Cent
2—Bearing next to bull pinions	19.5
1—Bearing on first reduction shaft nearest overhanging gear	15.5
1—Bearing on opposite end of first reduction shaft	5.5
1—Bearing on counter-shaft	5.5
1—Bearing on counter-shaft	5.5
2—Rear axle bearings	4.2
Plain bearing loss	55.7
Anti-friction bearing loss	1.85
Saving in favor of anti-friction bearings	*53.85

*=5.38 hp.

This amount is clear gain, and can be added directly to the drawbar and credit given for a net saving. In other words, a 20-hp. engine working at full load will, in eight out of ten places, deliver to the drawbar a little over 50 per cent more net power with anti-friction bearings than it would if plain bearings were used. This means a saving of a little over 50 per cent of the fuel required to do the same amount of drawbar work, or approximately 10 gal. of fuel per day's work of the plain-bearing type of tractor when under full load.

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In the particular tractor under consideration the additional cost of the anti-friction bearings amounted to approximately $30 at present market prices. This cost would be covered in fuel saving within a few days.

Few, if any, of us today would design an automobile or truck with plain bearings throughout the transmission and rear axle; still less would we think of putting a grease cup on each of these bearings and asking the user to crawl under and tighten these cups each hour.

When these fuel-saving and trouble-saving problems are presented to the prospective purchaser of a tractor it will be an easy matter for him to decide whether or not he should not pay 2 or 3 per cent more for a tractor that will save 50 per cent of his fuel on drawbar work and at the same time require lubrication once a week only instead of once an hour; and in addition to these advantages give him an opportunity to do his bit in conserving the world's gasoline supply.

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BUFFALO SECTION PAPER

WORM GEAR BRONZES

By W M CORSE*

Worm gearing is one of the oldest known mechanical movements. Its ability to give large reductions in speed in a simple and satisfactory way and by a compact mechanism has long been recognized. In the higher grades of worm gearing the worm is usually made of steel, specially treated to insure specific properties, and the worm wheel is made of bronze, also with specific composition and treatment. As this paper deals with the bronze element it will be of interest to learn something of the history of bronze, particularly as the oldest and best known worm-gear bronzes, the copper-tin, are the first about which we find any description in ancient literature.

In the excellent translation of De Re Metallica (published in 1556 by Georgius Agricola) which was made in 1912 by Herbert Clark Hoover, aided by his wife, some instructive notes are given on the history of bronze, from which I shall quote.

"It is possible to set up a description of the imaginary beginning of the 'bronze age' prior to recorded civilization, starting with the savage who accidentally built a fire on top of some easily reducible ore, and discovered metal in the ashes; but as this method has been pursued times out of number to no particular purpose, we will confine ourselves to a summary of such facts as we can assemble.

"Founders' hoards of the bronze age are scattered over Western Europe, and indicate that smelting was done in shallow pits with charcoal. With the Egyptians we find occasional inscriptions showing small furnaces with forced draft, in early cases with a blowpipe, but later—about 1500 B.C.—with bellows also. The crucible was apparently used by the Egyptians in secondary melting, such remains at Mt. Sinai probably dating before 2000 B.C. Charcoal was the universal fuel for smelting down to the eighteenth century.

"Bronze articles have been found in the prehistoric[1] remains in Egypt as early as the IV Dynasty (3800 to 4700 B.C., according to the authority adopted). The

*Manager Bronze Dept., The Titanium Alloy Manufacturing Company

[1]Page 402, Note 42—Historical note on copper smelting

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question of the origin of this bronze, whether from ores containing copper and tin or by alloying the two metals, is one of wide difference of opinion.

"To our mind the vast majority of ancient bronzes[3] must have been made from copper and tin mined and smelted independently."

Chemical analyses of these ancient bronzes show them to contain from 2 to 10 per cent of tin, so that bronze suitable for use in worm gearing has been known for thousands of years. It was not until the nineteenth century, however, that much bronze was used for worm gear wheels. Wood was almost universally used prior to this time, but the advent of the automobile focused the attention of the mechanical world on the use of a better material.

One of the most common uses for worm gear reductions, until the advent of the automobile truck, was for elevators and similar hoisting devices.

Automobile rear-axle worm gears were not developed by the hoisting device manufacturers in the United States, however, but were largely imported from England. The high efficiency of this type of gearing as now made has led to its adoption in many industrial fields.

The main difference between worm gearing used for stationary power transmission and that used for automobile trucks is in the compactness and accuracy of the machine work required on the worm gear unit itself.

In the less compact elevator mechanism the size of the gearing permits the use of lower grade material and less accurate workmanship. In the rear axle mechanism of the automobile truck, on the other hand, the space available and the mechanical efficiency demanded make it imperative that only high-grade materials and workmanship be employed.

Such a condition as has just been described lends itself profitably to scientific investigation.

The mechanical side of the problem has been described by H. Kerr Thomas in his book entitled "Worm Gearing." It has also been described before this Society by John Younger,[2] Cornelius T. Myers[4] and H. D. Church.[5]

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[3]Page 411, Note 52—Historical note on tin metallurgy

[2]Worm Drive Axles, S. A. E. TRANSACTIONS, 1914, Vol. 9, Part 1, pp. 215-223

[4]Manufacture of Worm Gearing by a New Process, S. A. E. TRANSACTIONS, 1915, Vol. 10, Part 1, pp. 152-171

[5]Refinements in Truck Design, S. A. E. TRANSACTIONS, 1916, Vol. 11, Part II, pp. 324-335

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The metallurgical side of the problem has not been as thoroughly discussed and it is in the hope of adding information to this part of the subject that the technical data in this paper are presented.

Hour-Glass and Straight Type Worm Gears

When worm gear axles were first used in this country we found two types of worm gearing about equally regarded. These were the hour-glass type and the straight type. The advocates of the hour-glass type specified a bronze having a higher compressive resistance than that for the straight type. As the hour-glass type of gearing bears on a considerable surface of the worm and wheel while the straight type bears on a line tangent to the pitch diameter of the gear, it is obvious that in the former type a greater total tooth pressure can be used without destroying the oil film than in the latter. This is evidently why a bronze wheel having a high compressive strength is required in the hour-glass type to withstand these increased pressures.

Phosphor Bronzes

The usual straight type of worm gearing requires a phosphor-bronze of about 88.7 parts of copper, 11 parts of tin and 0.3 parts of phosphorus. This alloy in chilled gears has the following physical properties:

Ultimate tensile strength, lb. per sq. in.	35,000-40,000
Yield point, lb. per sq. in.	22,000-25,000
Elongation in 2 in., per cent	6-10
Reduction of area, per cent	7-9
Specific gravity at 20 deg. cent	8.5
Brinell hardness number (500 kgm. load for 30 sec.)	75-85
Pattern maker's allowance for shrinkage, in. per ft.	0.125
Weight per cu. in., lb.	0.31
Compression, elastic limit, lb. per sq. in.	16 000
Coefficient of friction	0.0040
Modulus of elasticity	12,000,000 to 14,000,000
Resistance to impact, Fremont notched bar test (fractured section 7 by 10 mm.), kgm-meters	2 to 4
Endurance of alternation impact, Landgraf-Turner or Arnold test, alternations	150 to 300
Resistance to shear by impact, McAdam machine, ft-lb.	300 to 450

Copper-Tin-Zinc Alloys

A bronze for hour-glass worm gears composed of about 85 parts of copper, 13 parts of tin and 2 parts of zinc, having a Brinell hardness number of about 100, has been used to some extent.

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Chill-Cast Tin Bronze

The 11 per cent tin bronze was also used for the hour-glass type when chill cast. In fact it was in this type that the chill cast bronze was first used.

The fact that the hour-glass type of gearing requires accurate alignment in three planes while the straight

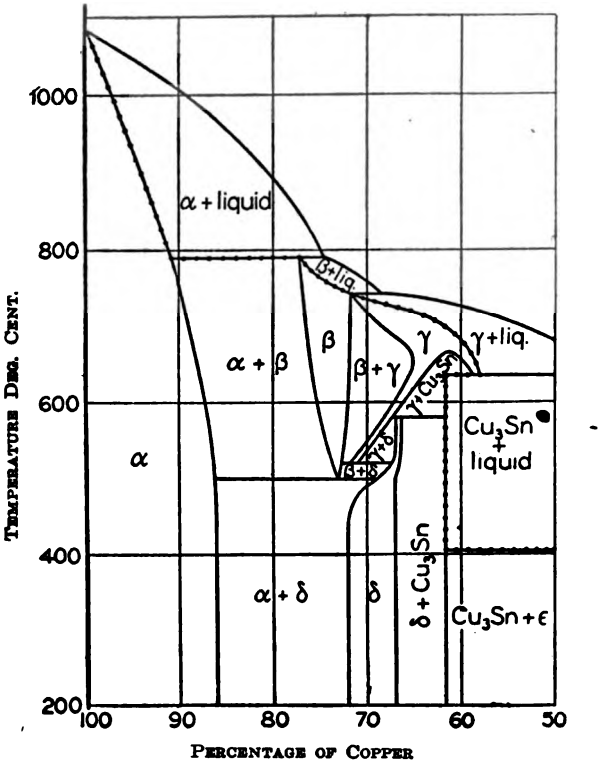

FIG. 1—THERMAL EQUILIBRIUM OR COOLING DIAGRAM OF COPPER-TIN ALLOYS

type requires accurate alignment in but two planes, has caused the latter to be adopted in automobile truck design. The hour-glass type is still largely used in electric passenger vehicles.

Copper-tin alloys to which phosphorus has been added are called phosphor-bronzes. When the percentage of copper is between 100 and 87 per cent the form of crystallization is the same. The temperature of the solidus of an alloy consisting of from 9 to 13 per cent of

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tin is about 200 deg. cent. lower than the temperature of the liquidus taken from the table of pyrometric data. (See Fig. 1*)

Effect of Phosphorus

The addition of phosphorus up to 1 per cent apparently increases this difference in temperature. This condition is common to many alloys and means that during the cooling process a solid portion of the alloy is in contact with a liquid portion during a temperature drop represented by the difference between the temperatures of the solidus and the liquidus. This temperature difference is constant for a given composition, when the rate of cooling is normal.

EXPLANATION OF ILLUSTRATIONS
(See insert)

Fig. 2—Bronze containing 11 per cent tin, cast in sand, etched with ammonia and hydrogen peroxide, followed by ferric chloride, showing light patches and streaks of eutectoid between the darkened alpha dentrites.

Fig. 3—Same alloy as shown in Fig. 2, with same etching but chill cast, showing only traces of eutectoid and much finer dentritic structure.

Fig. 4—Typical structure of sand-cast aluminum-bronze containing 10 per cent aluminum, etched with ferric chloride and hydrochloric acid.

Fig. 5—Same alloy as Fig. 7, with same etching, but after further heating to about 600 deg. cent., followed by slow cooling, showing a very fine grain similar to that found in tempered steel.

Fig. 6—Same alloy and etching as Fig. 4, but with a 20-fold greater magnification, showing duplex structure of the dark constituent.

Fig. 7—Structure of 10 per cent aluminum bronze, hardened by quenching in water from 900 deg. cent. and etched with ferric chloride and hydrochloric acid, showing acicular structure like martensite in steel.

ALLOY COOLING DIAGRAM

For the information of those unfamiliar with cooling curve diagrams, I would state that the solidus is the line bounding those regions within which alloys are completely solid, and the liquidus is the line bounding those regions of the diagram representing entirely fluid alloys.

In the diagram, Fig. 1, the line bounding the area γ is the solidus of completely solid alloys of copper and tin. The curve bounding the area marked "α plus liquid" in Fig. 1 is the liquidus of entirely fluid copper-tin alloys.

TIN SWEATING OR LIQUATION

In foundry practice the result of this difference in temperature causes a phenomenon known as liquation. It is commonly called tin sweating because, when the

*Fig. 1 is reproduced from "The Tensile Strength of Copper-Tin Alloys" by Shepherd and Upton, *Journal of Physical Chemistry* January, 1905, p. 446

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casting is poured in a sand mold numerous drops of a silvery metal ooze from the risers and top during the process of solidification. This silvery metal is not pure tin, but rather a copper-tin alloy containing about 22 per cent of tin and having a Brinell hardness number of about 240.

Examination of the bronze with a microscope reveals a structure like Fig. 2 in which the lake-like areas are the silvery colored material, known as eutectoid, which is distributed throughout the mass. As the weight of metal poured into the average automobile-truck gear mold is between 75 and 100 lb., the solidification process is relatively slow, thus allowing ample time for the formation of the eutectoid. The casting, therefore, contains large areas of this relatively hard material, which makes it more difficult to machine uniformly. This produces an uneven surface on the tooth of the gear with the attendant mechanical disadvantages of such a condition.

Chill Casting Produces Finer Grain

The most practical method for overcoming this unevenness of structure is to chill the liquid metal in the casting by mechanical means. Such chilling produces a finer grain in the metal throughout practically the entire cross-section of the gear rim and reduces the amount of eutectoid, or hard material, thus giving a more uniform alloy. This is clearly shown in Fig. 3.

The chilled phosphor bronze worm wheel has been proved to have many advantages over the ordinary sand cast article. Thousands of such gears in service have shown better wear and therefore longer life.

One of the essential properties of a gear bronze is the ability to burnish or cold-work its own surface. A high-grade bronze will increase in surface hardness from 30 to 50 numbers on the Brinell scale when burnished by machining or service operations.

Manganese Bronze

Manganese bronze possesses high strength but does not burnish well; it therefore does not make a satisfactory gear metal.

Aluminum Bronzes

The need for a strong gear bronze, however, in order to effect economies in the use of a relatively expensive material, has been recognized. This has led to the investigation of the aluminum bronze series of alloys; these are known to possess high strength, remarkable resist-

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ance to alternating stresses caused by shock and vibration, and resistance to abrasion and wear.

The composition with the best properties for gears is about 90 parts of copper and 10 parts of aluminum. Small percentages of iron added to the above formula give increased strength in tension and compression. The burnishing properties are found to be satisfactory, fully equalling or surpassing those of the phosphor bronzes.

Advantages of Aluminum Bronze for Worm Gears

The phenomenon of liquation present in the phosphor bronzes is found to be practically absent in the aluminum bronzes. This is shown in Fig. 1a.*

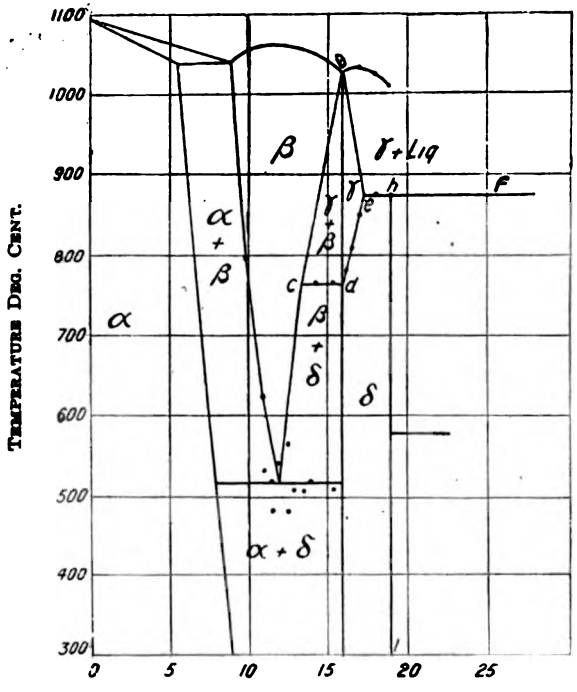

FIG. 1a—THERMAL EQUILIBRIUM OR COOLING DIAGRAM OF COPPER-ALUMINUM ALLOYS

Note small size of triangular area at upper left-hand corner of this diagram, as compared with large "alpha plus liquid" area in Fig. 1

*Fig. 1a is reproduced from "Some Experiments upon Copper-Aluminum Alloys," by J. H. Andrew, *Journal of the Institute of Metals*, March, 1915

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From the foundryman's standpoint the absence of liquation in these alloys means that if advisable the chilling process can be omitted and a bronze casting be produced that shows no hard spots with their attendant difficulties. Where it is desirable to partly cast the teeth in the gear, the aluminum bronzes, because of the absence of the hard spots in the sand cast material, offer greater possibilities than the phosphor-bronzes.

In addition, the 10 per cent aluminum bronzes can be heat-treated, giving a metal with 25 per cent increase in tensile strength, 100 per cent in yield point and 50 per cent in hardness without any increase in machining.

In certain designs these improved properties may be found very beneficial. Figs. 4 and 6 show the structure of aluminum bronze under the microscope. Fig. 7 shows the structure of quenched aluminum bronze and Fig. 5 that of the heat-treated alloy.

The 10 per cent aluminum bronze containing 1 per cent of iron has the following physical properties:

Ultimate tensile strength, lb. per sq. in.	65,000-80 000
Yield point, lb. per sq. in.	23,000-28 000
Elongation in 2 in., per cent	20-30
Reduction of area, per cent	21-29
Specific gravity at 20 deg. cent.	7.5
Brinell hardness number (500 kgm. load for 30 sec.)	92-100
Pattern maker's allowance for shrinkage, in. per ft.	0.22
Weight per cu. in., lb.	0.27
Compression, elastic limit, lb. per sq. in.	19.000
Coefficient of friction	0 0025
Modulus of elasticity	15,000,000-18,000,000
Resistance to impact, Fremont notched bar test (fractured section 7 by 10 mm.), kgm meters	7 to 10
Endurance of alternating impact, Landgraf-Turner or Arnold test, alternations	3500 to 5500
Resistance to shear by impact, McAdam machine, ft.-lb.	750 to 850

Addition of Iron to Aluminum Bronze

A 10 per cent aluminum bronze containing 4 per cent iron, first described* by G. F. Comstock and the author in 1916, shows results superior to the above and bids fair to supersede it.

Economy of Material with Aluminum Bronze

Recent tests of worm gear axles by the U. S. Army on the Mexican border demonstrated the need of a gear bronze with a high resistance to shock. A comparison of

*"Some Copper-Aluminum-Iron Alloys," Corse and Comstock. *Trans. Amer. Inst. of Metals*, Vol. X, pp. 119-132

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the results obtained by the Fremont and Landgraf-Turner tests shows the great superiority of the aluminum bronzes in this respect. With the greater strength and toughness of aluminum bronze as compared with phosphor bronze, and its decreased weight per cubic inch, the designer can effect economies that total from 30 to 50 per cent in material.

The manufacturing difficulties formerly present with aluminum bronze have been overcome so that worm gear wheel castings of this material are now being made in quantities with as low a percentage of rejections by the purchaser as with phosphor bronze.

In the new field of tractor designing, aluminum bronze should be given careful consideration on account of the economies possible to effect through its use in worm gear castings and other heavy-duty bronze parts.

I do not wish to be understood as recommending one class of alloys over the other for all gear castings. Undoubtedly the mechanical phases of the problem have a direct bearing on the selection of the proper gear bronze for the job.

The hardness of the steel worm, its machine finish and the accuracy of finish on the worm wheel teeth have a direct bearing on the proper selection. These conditions are best determined by the designing engineer. It is my desire to give the metallurgical facts in each instance and leave the final selection to the engineer because of his experience with the finished worm gear mechanism. I do wish to point out, however, that the bronze is fully as important as the other parts of the mechanism and should be as carefully made and controlled.

With the idea of adding to present information on the subject, I have attempted to present here such metallurgical data as would seem to be pertinent.

I wish to acknowledge my indebtedness to Hugh R. Corse for many valuable suggestions relating to the subject matter of the paper and to George F. Comstock for his aid in preparing the microphotographs and tables.

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CLEVELAND SECTION PAPERS

EFFECTS OF LOW TEMPERATURE ON STARTING

By O W A OETTING*

The starting troubles encountered because of cold weather conditions are most exasperating to the car user, but in the majority of cases are avoidable if all the effects of low temperatures have been anticipated by the automobile designer. The proper application of the electric starter and storage battery for use on passenger cars, motor trucks and tractors presents problems to the engineer which, if properly solved, will prevent some of the troubles which have occurred frequently in the past.

No apparatus is required to operate under such a variety of conditions as the automobile engine and its accessories. The range of temperature from the heat of summer to the freezing cold of winter is one of the conditions that must be appreciated in making the proper application of the storage battery for this service. Storage batteries that will operate satisfactorily under these extreme conditions can be made, but engineers must take into consideration certain existing facts. In cold weather, the capacity and the voltage of a battery are less than at normal temperatures. Likewise, it is much more difficult to start an engine because of the effects of these cold temperatures on the lubrication and also on the carburetion of the fuel. In addition to these facts we find that the efficiencies of electric starting motors are considerably less at the low cranking speeds encountered under these cold weather conditions. *The factor that must be used in determining the size of the storage battery is the power required to start the engine under cold weather conditions.*

Considerable data on cold weather starting have been secured. It is the purpose of this paper, first, to give a short résumé of these tests and, second, to summarize these results so as to predict, if possible, the size of storage battery that is required for starting any engine, giving the approximate ampere-hour battery capacity in terms of the piston displacement of the engine.

*Engineer, Willard Storage Battery Company

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Battery Sizes

Heretofore, not enough attention has been given to the size of storage battery required on a car and almost invariably a small size of battery has been used when a larger battery would have given better service. This haphazard application of the battery shows great inconsistencies when a plot (Fig. 1) is made of the battery sizes against engine displacements. The data for this plot are taken from a list of all the 1917 passenger cars and are no doubt representative of present-day

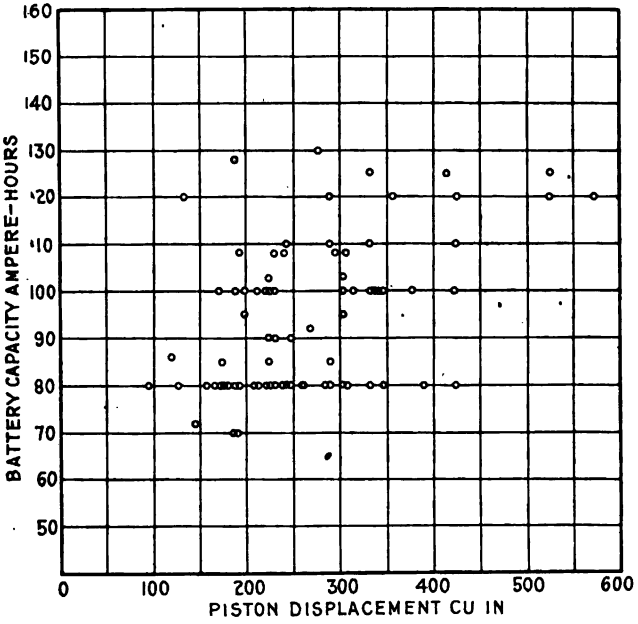

Fig. 1—Comparison of Size of Storage Battery with Engine Displacement on 1917 Passenger Cars

practice. It can be seen that the 80 ampere-hour battery is used on cars having piston displacements from 95 to 425 cu. in. On the other hand, the 100 ampere-hour battery is used on cars with displacements ranging from 170 to 425 cu. in. Undoubtedly, cars with piston displacements over 200 cu. in. should be equipped with a larger battery than 80 ampere-hour and those with displacements greater than 250 cu. in. should have a battery larger than 100 ampere-hour. The foregoing

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plot is true, of course, only if the gear ratios between the starter and engine are the same. On present-day makes of cars this is practically so, inasmuch as the majority of the cars are equipped with the single-gear reduction with ratios approximately equal to 10 to 1.

With the purpose in mind of securing some factor that will determine definitely the size of battery required for every size of car, engines were tested with their starting motors and batteries in a large refrigerator to find the actual power required to turn over the engine at various temperatures. The exact minimum of temperature at

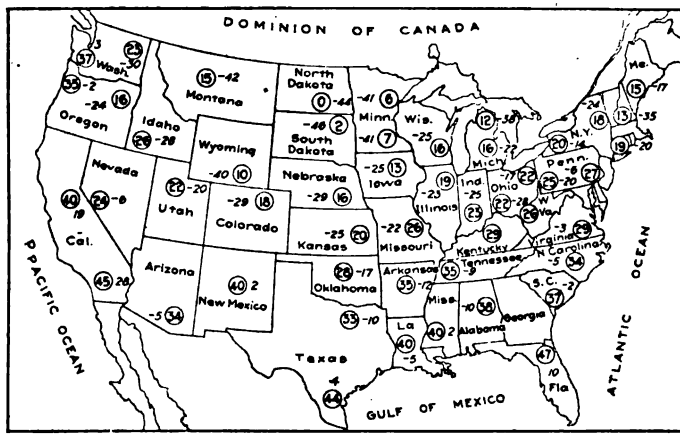

FIG. 2—AVERAGE TEMPERATURES IN VARIOUS LOCALITIES THROUGHOUT U. S. DURING THREE WINTER MONTHS

Note: Small numbers beside circles give absolute minimum temperatures.

which the engine should start promptly is of course debatable.

AVERAGE LOW TEMPERATURES

Fig. 2 shows a chart which was compiled from data on temperatures obtained by the U. S. Weather Bureau. The figures within the circles are averages observed during the three winter months for a long series of years at various stations throughout the United States. The small figures beside the circles give the absolute minimum temperatures in the same localities during this period of time. It will be noted that only three states have an average temperature lower than 10 deg. fahr. If then we take 10 deg. fahr. as a cold weather standard

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and design the engine and all the automobile accessories to operate properly at this temperature, little trouble should be experienced during the winter months and satisfactory service from the apparatus will be assured.

One of the effects of low temperature on a storage battery is to lower the terminal voltage, especially at the high rates of discharge that are required by the starting motor. Fig. 3 shows the "five-second" voltage curves of a 100 ampere-hour battery at 80 deg. fahr. and at 10 deg. fahr.

By the term "five-second voltage" is meant the voltage that a battery will give at the end of five seconds at a certain discharge rate of current. In other words, the battery represented by Fig. 3 will have a voltage of 4.82 after five seconds discharge at 400 amperes at a temperature of 80 deg. fahr. It will be seen from this curve that this battery would not be suitable for an engine with a starter requiring 300 amperes at 4.5 volts when the tem-

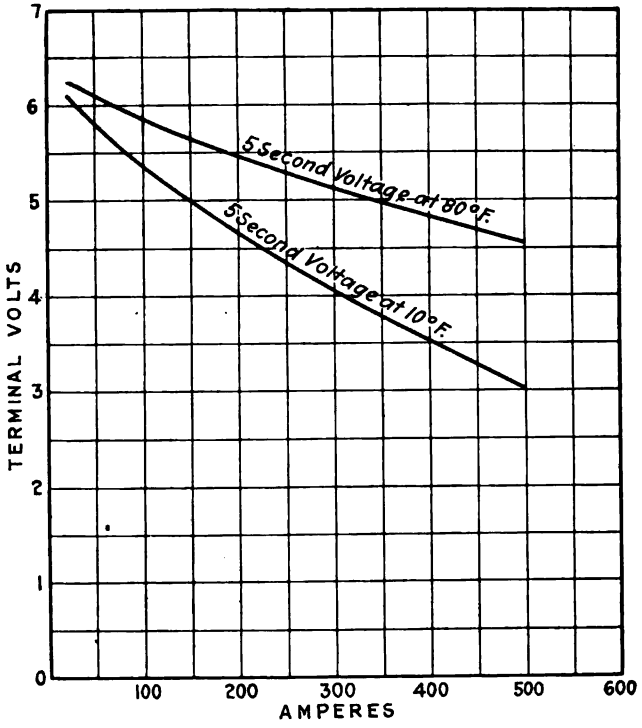

FIG. 3—COMPARISON OF VOLTAGE CHARACTERISTICS OF 100 AMPERE-HOUR STORAGE BATTERY AT 80 DEG. AND 10 DEG. FAHR.

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perature was 10 deg. fahr. or lower. Curves such as these should be used in all starting motor designs.

Fig. 4 is a picture of a refrigerating machine and cooling chamber used for testing an automobile engine with its starter and battery at low temperatures. This refrigerator can be cooled to a temperature as low as 30 deg. The engine to be tested is placed within the box

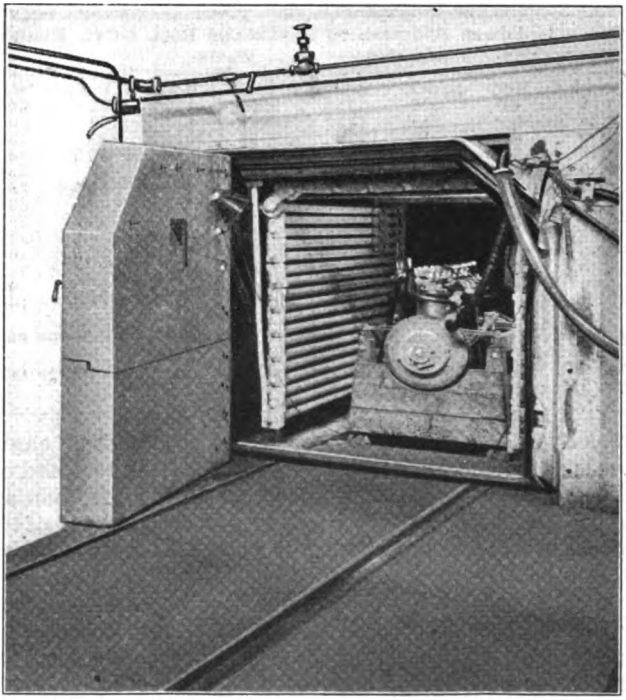

FIG. 4—REFRIGERATOR USED FOR TESTING ENGINES FOR COLD WEATHER STARTING CHARACTERISTICS

and wires are brought out from the battery and starter to read the electrical input required to start the engine at low temperatures. A thermometer and some anti-freeze solution are placed in the cooling chamber of the engine and the temperatures can be observed from the outside of the box by means of a small observation door in the main door of the box. The revolutions of the engine are obtained by means of an electrical contactor, which rings a bell outside of the refrigerator. After the starting data are obtained at the normal room temperature, the box is slowly cooled to any desired temperature and the starting characteristics of the engine at this temperature are then observed.

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Test Results

Table I gives the results of a test on a four-cylinder engine having a piston displacement of 171 cu. in. The values in this table give the power that is required to break away the engine and then roll it at a certain speed and temperature. These results are plotted in Fig. 5, the curves showing the average power used for the

TABLE I—POWER REQUIRED TO BREAK AND ROLL 4-CYL. ENGINE

Motor Volts		Amperes		Watts		Speed,	Temp.,
Break	Roll	Break	Roll	Break	Roll	R.P.M.	°F.
4.75	4.0	200	175	950	700	61	60
5.20	5.0	260	175	1350	875	130	60
5.90	5.75	210	180	1240	1035	162	60
4.15	3.60	245	275	1020	990	42.5	30
4.60	4.25	280	270	1285	1150	57	30
4.70	4.25	260	280	1220	1190	65.5	30
4.50	4.25	275	285	1240	1210	66.5	30
5.60	5.10	260	265	1510	1350	83.5	30
3.90	3.4	310	280	1210	950	24.5	10
3.70	4.2	290	310	1075	1300	31.5	10
3.90	4.2	315	300	1230	1260	33	10
5.80	4.5	335	330	1950	1485	43	10
6.10	5.5	380	390	2320	2140	71.5	10

Figures in columns headed "Break" are the instantaneous readings taken the moment the circuit closed.

Figures in columns headed "Roll" are the average readings taken while the engine was rolling.

break-away and roll of the engine at various engine speeds and temperatures. The engine in this test had one cylinder which required about 50 amperes more than any one of the other three cylinders during the roll of the crankshaft. This, however, is the service that is demanded of a new storage battery on a car, as the engines are often not "run in" sufficiently before shipment is made by the car manufacturer. A 90 ampere-hour battery at a temperature of 10 deg. fahr. turned over this engine at a temperature of 8 deg. fahr. at 30 r.p.m.

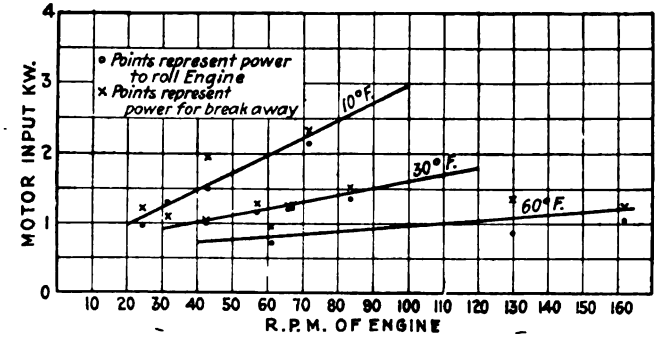

FIG. 5—POWER REQUIRED AT VARIOUS TEMPERATURES BY STARTING MOTOR ON A 4-CYLINDER ENGINE OF 171 CU. IN. DISPLACEMENT

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A considerable variation in the power demand from the storage battery is caused by poor design and application of the starting motor. Fig. 6 shows curves of power used by four different types of starters on a six-cylinder engine. The light weight starter *D* required about 1 kw. of power more than the starter *A*, in which the proportions of the various parts were more liberal. These curves

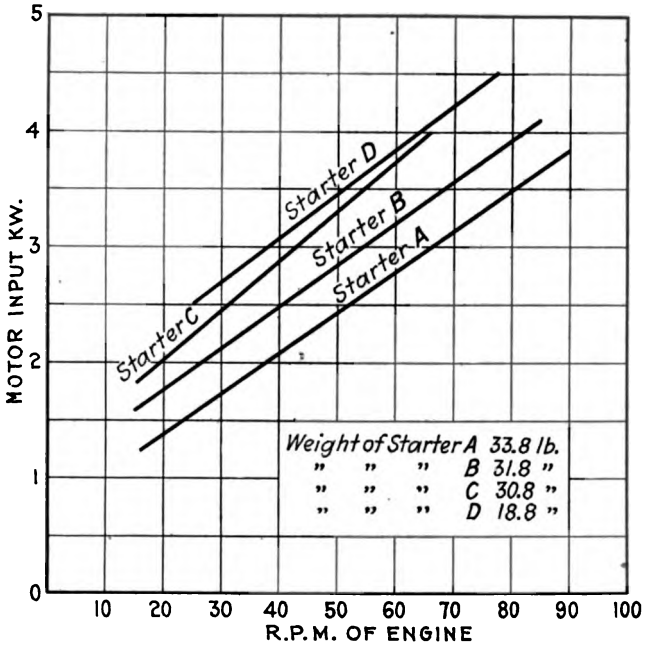

FIG. 6—POWER REQUIRED AT 10 DEG. FAHR. BY DIFFERENT TYPES OF STARTERS ON A 6-CYLINDER ENGINE OF 268 CU. IN. DISPLACEMENT

indicate that the tendency to a lighter and cheaper starter will necessitate the use of a larger storage battery. The automobile engineer must therefore decide where a compromise between the battery and starter will give the best economy and service. In the case of the engine in question, starter *B* was used. At 30 deg. fahr. this starter with a 120 ampere-hour battery rolled the engine at a speed of 30 r.p.m.

In Fig. 7 corresponding curves are shown on an eight-cylinder engine having a displacement of 332 cu. in. This engine was made by a well known engine manufacturer and the fittings of all the parts were well made. The results on this test were therefore lower in proportion to

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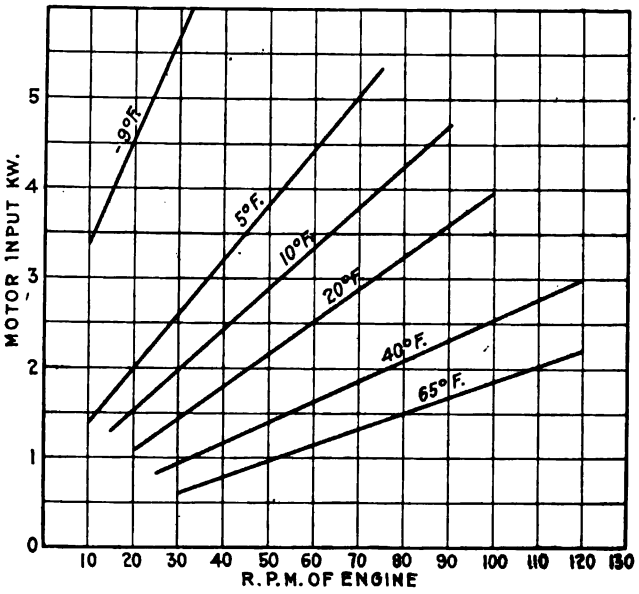

FIG. 7—POWER REQUIRED AT VARIOUS TEMPERATURES BY STARTING MOTOR ON AN 8-CYLINDER ENGINE OF 332 CU. IN. DISPLACEMENT

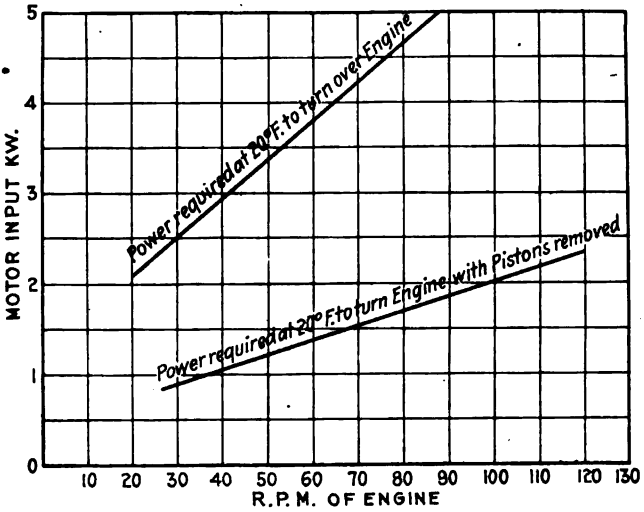

FIG. 8—COMPARISON OF POWER REQUIRED TO TURN A 6-CYLINDER ENGINE OF 303 CU. IN. DISPLACEMENT BEFORE AND AFTER REMOVING PISTONS AND CONNECTING-RODS

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the size of the engine than on some of the other engines tested at these cold temperatures.

A bad feature in the design of bearings was brought out in one of the cold temperature tests during these investigations. The bearing caps on one of the engines tested were made of aluminum, and at the low temperature these bearings were found to be shrunk tight on the shaft, due to the difference in the coefficients of expansion of the two metals. During the test, it was observed that this engine required an unusually large amount of power to start it. The pistons, connecting-rods and camshaft drive were therefore removed; the power required to turn the crankshaft and flywheel was found to be considerable. Fig. 8 shows that the power required for this latter condition was about one-third of the total power required to break-away and turn the engine without the pistons removed. At 6 deg. fahr. with the pistons removed and 2.8 kw. supplied to the starter it was impossible to pry the engine loose with a 10-in. lever in the teeth of the flywheel. The results of this test gave values that were about 100 per cent higher than the average of all the engines tested under these cold weather conditions.

Summary of Tests

Other engines were tested in addition to those mentioned in the foregoing and all the results were averaged so as to determine, if possible, the power required to start any size of engine. It was found that it is hardly possible to lay down a hard-and-fast rule for this purpose, but a close approximation is possible. Average results on the engines tested differed for various reasons, such as variation in compression, difference in viscosity of the lubricating oils at low temperatures, variation in refinement in the fittings of the engine, and difference in design of the starting motors.

Fig. 9 and Fig. 10 give a summary of the power required to start the various sizes of engines at normal temperatures and 10 deg. fahr. respectively. These curves are plotted with the axes of motor input against the piston displacement of the engine. It would have been more logical perhaps to plot the power used against the area of the rubbing surface of the piston walls, but this relation did not average the points any better than the one shown in the curve. For convenience, then, the co-ordinates used in the curves were chosen.

Having ascertained the power required to start the engine, but one more step is necessary to determine

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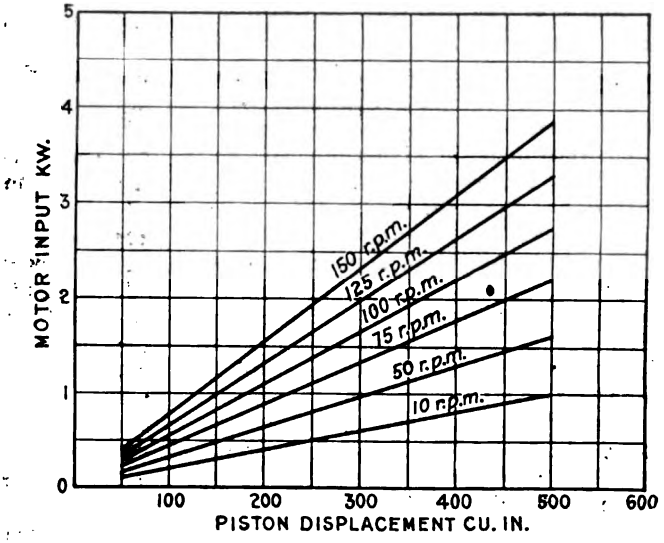

FIG. 9—APPROXIMATE POWER REQUIRED TO BREAK-AWAY AND TURN OVER VARIOUS SIZES OF ENGINES AT NORMAL TEMPERATURES

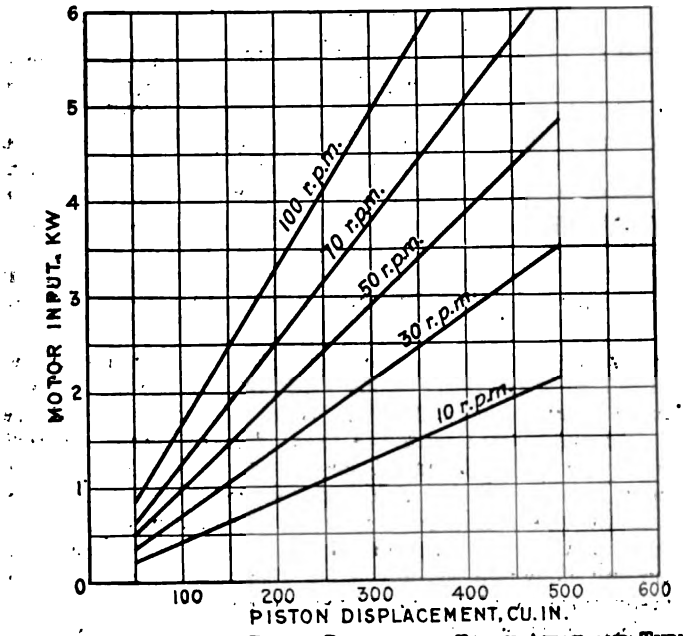

FIG. 10—APPROXIMATE POWER REQUIRED TO BREAK-AWAY AND TURN OVER VARIOUS SIZES OF ENGINES AT 10 DEG. FAHR.

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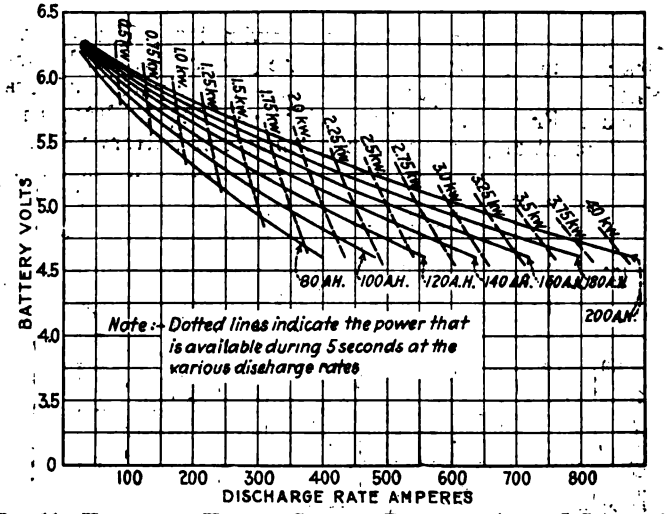

FIG. 11—VOLTAGE OF VARIOUS SIZES OF BATTERIES AFTER 5 SECONDS' DISCHARGE AT 80 DEG. FAHR.

the approximate size of storage battery that is needed for this service. Fig. 11 and Fig. 12 show the "five-second voltage" curves of various sizes of batteries at 80 deg. fahr. and 10 deg. fahr. respectively. The power that is available for five seconds from any of these batteries is the product of the volts and the amperes at any

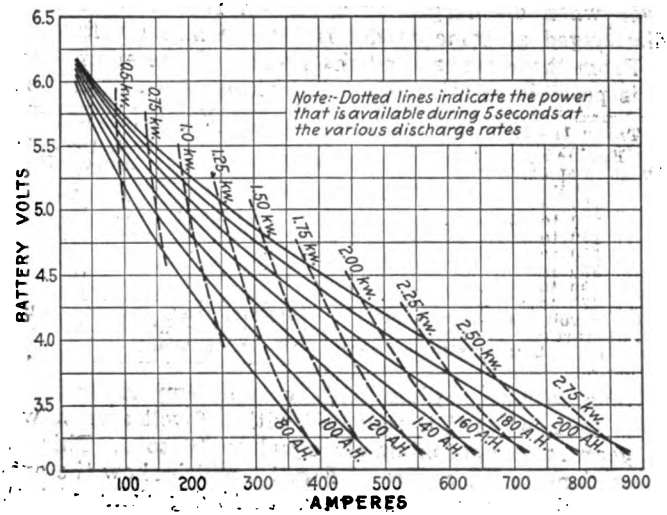

FIG. 12—VOLTAGE OF VARIOUS SIZES OF BATTERIES AFTER 5 SECONDS' DISCHARGE AT 10 DEG. FAHR.

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of the intersections of these coordinates. The dotted lines on these curves represent the power that can be obtained from any of these batteries for a period of five seconds. From these curves it is possible to determine the amount of power that a given size of battery will de-

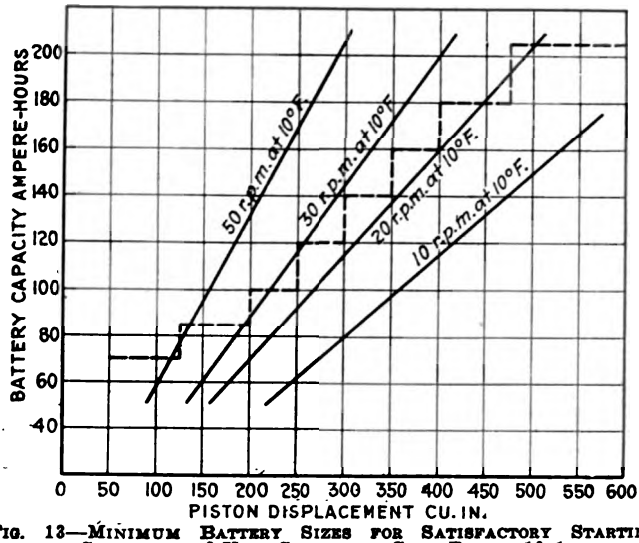

FIG. 13—MINIMUM BATTERY SIZES FOR SATISFACTORY STARTING SERVICE FOR 6-VOLT STARTERS. GEAR RATIOS 10:1

liver when discharged at a certain rate of current or discharged at some predetermined terminal voltage.

From these curves and the results of the engine tests it is possible to estimate the cranking speed of any size

TABLE II—MINIMUM BATTERY SIZES FOR STARTING SYSTEMS

Engine Size Cu. In. Displacement	Battery Size Ampere-Hours
Up to 125	70
125 to 200	85
200 to 250	100
250 to 300	120
300 to 350	140
350 to 400	160
400 to 475	180
475 to 600	205

For 6-volt systems with gear ratio approximately equal to 10 to 1. For starting systems of higher voltage or gear ratios greater than 10 to 1, the battery size can be reduced proportionally.

of engine with a given battery at a temperature of 10 deg. fahr. when it is equipped with a 6-volt starter with a gear ratio approximately equal to 10 to 1. This has been done in the curves of Fig. 13, which show the average results of the engine speeds at 10 deg. fahr. with the size of storage battery plotted against the size of the engine. Table II gives the minimum size of battery that

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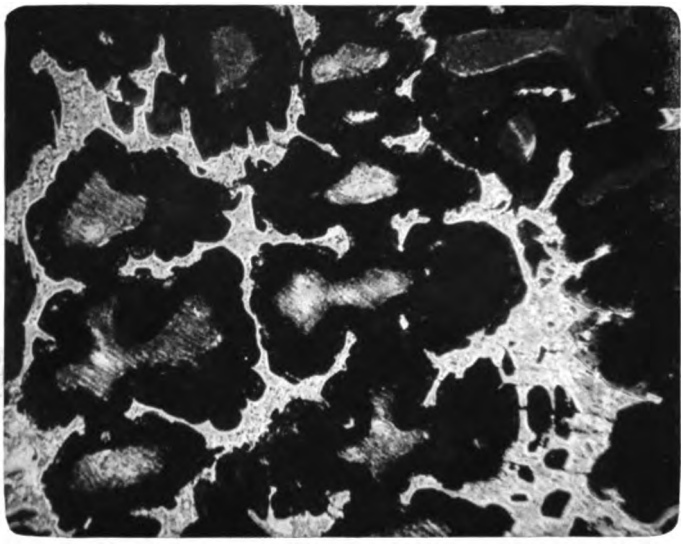

FIG. 2—COPPER-TIN ALLOY CAST IN SAND
Magnification—200 Diameters

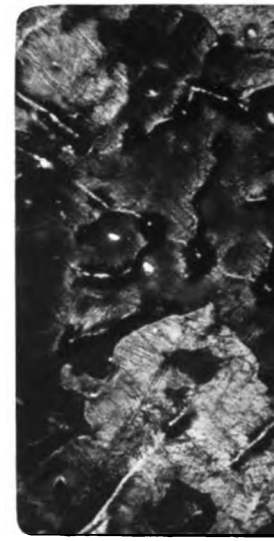

FIG. 3—SAME ALLO
Magnific

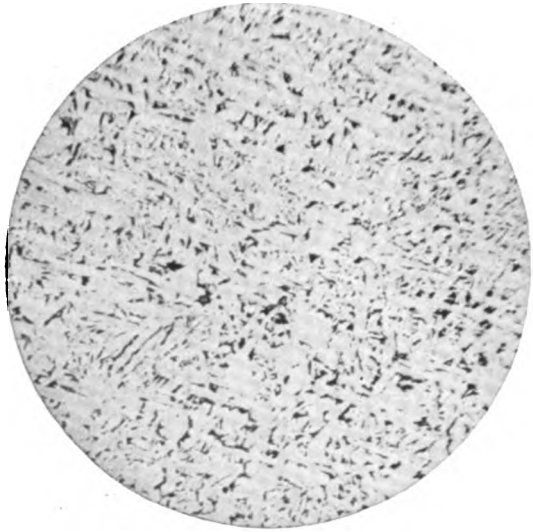

FIG. 5—SAME ALLOY AS IN FIG. 7, BUT REHEATED AND COOLED SLOWLY
Magnification—200 Diameters

FIG. 6—SAME ALI
Magnification—

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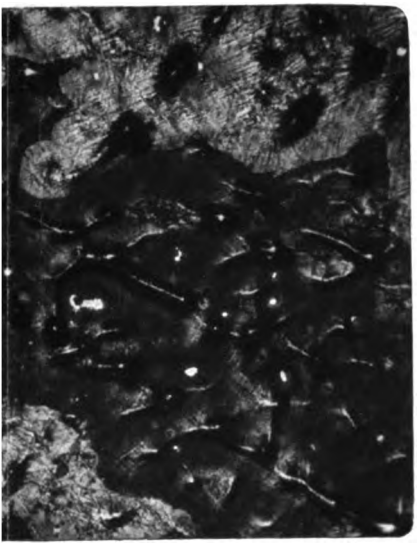
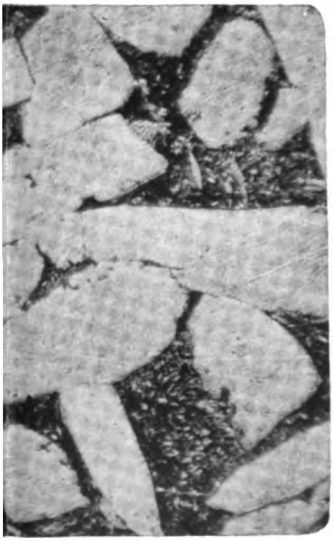


OY AS IN FIG. 2, BUT CHILL CAST
ication—200 *Diameters*

FIG. 4—ALUMINUM-BRONZE ALLOY CAST IN SAND
Magnification—20 *Diameters*



LOY AS IN FIG. 4
100 *Diameters*

FIG. 7—ALUMINUM-BRONZE HARDENED BY QUENCHING IN WATER
Magnification—200 *Diameters*

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is satisfactory for use on a 6-volt starting system when the gear ratio is approximately equal to 10 to 1. The basis for these data is the power that is required by the starter at 10 deg. fahr. as shown in Fig. 10 and also that power which is available at this temperature from the battery when the rate of discharge is about four times its twenty-minute rate. No allowance was made for the drop in the leads or starting switch between the starter and the storage battery. This drop varies in the different applications from 0.2 to 0.5 volts and should be made as small as possible, so as to utilize the maximum power that is available from the battery.

The values given in Table II have been indicated in Fig. 13 by the dotted lines. It will be noted that the battery sizes given in the table lie between the cranking speeds of 20 and 50 r.p.m. at 10 deg. There are diverse opinions in regard to the minimum speed for successful starting, a condition which necessarily depends not only on the electrical system but also on the ignition, lubrication and carburetion of the fuel. The battery capacities recommended in Table II are the *minimum* sizes that will give satisfactory starting service at 10 deg. fahr. in accordance with the average results of all these low temperature investigations. Some engines may be found that will require larger batteries than these sizes and in such cases greater capacities should undoubtedly be used.

One of the features that was brought out forcefully in these cold temperature tests was the effect of the cold on the viscosity of the lubricating oils. As the temperature decreases, the viscosity increases and causes a much greater resistance to turning. One certain grade of oil became a solid mass at these cold temperatures. A medium grade of oil in another test was found to have the consistency of extra heavy oil when cooled to 10 deg. fahr. Therefore, a light or a medium grade of oil of a quality that will not lose its lubricating properties at 10 deg. fahr. or hinder the successful operation of the starting system should be used on cars during the winter months.

Another of the features noted in these investigations, namely the effect of low temperatures on the ignition system, showed clearly one of the reasons for the adoption of battery ignition in place of magneto ignition. The cranking speed was seldom rapid enough to fire the fuel in the cylinders when the magneto was used. However, when a battery of sufficient capacity is used to turn the engine over successfully, the battery voltage will

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always be high enough to secure combustion in an engine equipped with a well-designed battery ignition system.

The grade of fuel and also its carburetion have a great influence on successful starting at low temperatures. The installation of a good electrical system will be utterly negative in results if the grade of fuel or poor carburetion keeps the engine from firing. If the electrical system turns over a cold engine at some reasonably high speed, it should be unnecessary for the car user to be put to the inconvenience of heating the carbureter and manifold of the engine to cause the ignition of the fuel.

Author's Conclusions

1 The size of a storage battery for starter service is determined by the power required to start and roll an engine at some reasonably cold temperature (10 deg. fahr.) at a sufficient speed to cause ignition.

2 Starting troubles may be due to lack of battery capacity, improper application or lack of size of the starter, viscosity of lubricant at low temperatures, faulty ignition, poor carburetion, or an inferior grade of fuel.

3 A good design of engine and its accessories should take into consideration all of these causes and provide for satisfactory starting at a temperature no higher than 10 deg. fahr.

4 At 10 deg. fahr, the available battery capacity is approximately 50 per cent of the normal capacity and the voltage at this temperature also is reduced to a considerable degree.

5 At 10 deg. fahr. the efficiencies of the starting motors vary between 10 and 40 per cent for the low cranking speeds encountered. A design of motor should be used which gives better starting torques and higher efficiencies at these low speeds, even if it is necessary to sacrifice the speed at which the motor spins the engine under normal temperatures.

6 A convenient double-reduction gearing between the starter and the engine flywheel can be made that will give better starting characteristics under cold weather conditions than a single-gear reduction.

7 A good quality of lubricating oil and a good grade of fuel are requisites for satisfactory starting.

8 As there is a wide divergence in the design and manufacture of the various automotive engines, each application should be tested separately to determine the power that is required to start it under cold weather con-

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ditions. In the absence of such data, the approximations derived in this paper should be useful.

THE DISCUSSION

A MEMBER:—In the case of lubricant actually frozen or solidified, was the power required to roll the engine after it was broken loose found to be less or greater than when the oil was not frozen?

MR. OETTING:—I have no exact data on that point. More power would be required after it was frozen, for it would probably have to run for a little while to bring up the temperature.

A MEMBER:—Would the average starter break it loose when frozen that way?

MR. OETTING:—We made tests of different grades of lubricating oils in test tubes, our one object being to find out if any engines were using a grade of oil that would solidify. I remember removing the thermometer from one test tube of oil that had stood over night, and a hole remained in the oil.

A MEMBER:—Was a rule formulated for picking out a good oil?

MR. OETTING:—No, it was not. There should be definite specifications written by automobile engineers for testing oils, and recommendations made to car users of oils suitable for successful service.

CHAIRMAN STRICKLAND:—One takes chances in putting a light oil into a high-powered engine to take care of freezing temperatures, especially when traveling rapidly after starting the engine. We have heard a good deal the last two winters about these conditions. We have been getting water in the crankcase and thinning the oil. In some cases I believe that water circulates with the oil, and, when stopping, more or less moisture is left in the bearings; that freezes, and is partly responsible for "sewing up" the engines.

JOHN MCGEORGE (M. S. A. E.):—The burden of the paper seems to be that low temperature results in freezing the engine, and that is the fault of the oil in the engine. Is it not a fact that the battery itself causes some trouble? Does not low temperature have some effect on the action of the battery?

MR. OETTING:—Yes. The fact was mentioned that at 10 deg. fahr. the battery capacity is 50 per cent less than at normal temperature. It is an inherent quality of the battery that cannot be changed in any way. We must design with that in mind.

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A MEMBER:—When using it the engine has approximately the same friction in cold as in warm weather, after it has been broken loose. That is one of its peculiarities. I had occasion to try that recently on an engine that had been standing in the cold, unused, for about one month; after I broke it loose it cranked as easily as in summer time, and also started properly. To break it loose I just pulled on the starting crank. It was not very hard. That engine has had a graphite lubricant in it from the time it was tested, about six years ago; the valves have never been ground, and the compression is as good as ever. Except for the last two or three months it has run every day for five years. The graphite particles in the lubricant used are so fine that they will go through a porcelain filter without clogging. If the cylinder is cracked a little line will be noticed where the lubricant has gone into the iron after running an engine with it a few thousand miles.

A MEMBER:—Does the temperature of the electrolyte make any difference, and does it have the same effect at all densities?

MR. OETTING:—At about 1.300 density of the acid electrolyte the freezing point is around —90 deg., and as the density becomes less the freezing point is raised. It is best to keep the battery charged so it will not freeze.

A MEMBER:—Suppose the density is down to one-half of normal. There will be a certain discharge from the battery if the motor is started at a temperature of, say, 50 deg.; now, would there be the same percentage of drop at 10 deg.?

MR. OETTING:—I think the percentage would be the same, but it is dangerous to let the battery go down in cold weather.

Larger Batteries Advisable

A MEMBER:—Should car manufacturers use larger batteries?

MR. OETTING:—The majority of them ought to have larger batteries. As Fig. 1 of my paper showed, no relation between engine size and battery size seems to have been adopted. There should be a definite relation, and I would say 75 per cent ought to have larger batteries.

A MEMBER:—In view of the fact that manufacturers will not pay more money, would it not be feasible to use thinner plates?

MR. OETTING:—Thinner plates cut down life. I think the majority of manufacturers put in batteries that usually last 18 to 24 months.

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CHAIRMAN STRICKLAND:—It seems to me that the curves used in the paper were based on average conditions. The first chart shows the sizes of batteries on all the cars, and of course it could be said that those under the average line should have larger batteries and those over the line should have smaller batteries. Are there other causes for the deterioration of batteries than those mentioned?

T. R. COOK (M. S. A. E.):—I do not know that one can name them. A battery simply runs down after giving out so much current. The result of deterioration of the plate is overheating in summer time. With a large battery installed one can charge sufficiently to take care of conditions in winter time, and yet not overheat the battery in the summer time. Records from our service stations show that the average battery life is about 24 months. That applies to batteries used on different makes of cars. There is one particular car that uses an oversize battery, and these have a life of 30 months, while cars using smaller batteries give out in 12 or 16 months. I think it is safe to say that the greatest trouble is overheating the smaller batteries in summer. With a small battery, the charging necessary to take care of the cranking and lighting in winter is of such amount that in summer the battery temperature would increase to 140 deg. or 160 deg. This results in rapid shedding of the plates.

A MEMBER:—I do not believe the size of the battery makes much difference; for instance, starting out at 10 deg., and making several calls, taking over 15 seconds time to start the engine, how long would 150 amperes last? The engine would not get warm enough to run over five miles an hour. What difference would size make if the engine would not start after one hour's use?

Motor Efficiency

CHAIRMAN STRICKLAND:—That point is well taken. I think the causes for the running down of batteries are various. The best and largest batteries will give out. One conclusion seems remarkable to me, because it differs from our original basis. In reference to the design of the motor for efficiency, a question came up as to whether to have it efficient at high-cranking speed or at low-cranking speed. The basis on which we formerly worked was to have it efficient at the standard cranking speed, which at that time was around 125 r.p.m. under ordinary conditions. Our assumption was based on the belief that batteries were run down because of other troubles—ignition or carburetion. Our claim was that

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the more efficient we made the motor at that speed the longer we could keep on spinning it, and finally get started; or, in other words, there would be more life in the battery. Now, these cold spells, 10 deg. and lower, where tremendous current is required for low-speed cranking, do not occur very often, and I do not consider it policy to proportion the motor to have its best efficiency at low speeds. The author of the paper differs from that view, and I would like to hear other reasons. I think some of the trouble lies in the poor upkeep of the generator; for example, brushes are allowed to wear down.

Mr. COOK:—In the winter time more lighting is required than in summer, and it is therefore desirable to take out as little current as possible. Consequently, better motor efficiency at low speeds is of advantage.

CHAIRMAN STRICKLAND:—I did not mean to advocate maximum motor efficiency for summer time; neither do I think that the author meant at 10 deg. He referred to low speeds; what speeds does he mean?

MR. OETTING:—Until recently, we were ignorant of the fact that cranking speeds of 10 to 20 r.p.m. were obtained at 10 deg. fahr. The starter should be designed to have higher efficiency when it is most needed. If a car can be started at 10 deg. it can be started easily at 70 to 80 deg. The point I make in the paper is that the battery size is determined by the power necessary to crank the engine at 10 deg., and what will take care of it at 10 deg. will take care of it at any temperature. As shown on the chart, I took 10 deg. as an average low temperature. Many places get as cold as —30 or —40 deg., but those are extreme conditions that it would not be right to ask manufacturers to meet.

Evaporation of Water

A. J. SCAIFE (M. S. A. E.):—With a small battery the water evaporates very rapidly in summer, which I imagine is very hard on the plates. I know of a twelve-volt battery that during summer would go for a month before the water would get below the top of the plates. It was a battery of ample capacity, and as it was large enough to take care of the output of the generator it never became very warm, and the water was not evaporated.

CHAIRMAN STRICKLAND:—I believe the battery people advise filling frequently in summer time.

MR. OETTING:—Fill the battery once a week for constant running in summer.

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A MEMBER:—What would be the effect of putting oil over the surface of the battery?

MR. COOK:—That was done years ago, but it did not have the desired effect. When trying to put in more current than the battery can take the battery starts to give off gas, and the electrolyte level drops.

CHAIRMAN STRICKLAND:—How long was the engine left in the cold room before making these tests?

MR. OETTING:—To get an engine down to zero, or 10 deg., to insure a uniform temperature throughout, it had to be left in the box over night. Anti-freeze has been placed in the cooling chamber to aid in this.

CHAIRMAN STRICKLAND:—I have found also that a car left in the cold two days is much worse than when left for one day. Probably it reaches a certain point where it will stand still, and I know leaving it from Saturday to Monday makes a decided difference.

A MEMBER:—The idea, apparently, is to install a very large storage battery to start the motor. I have a 12-volt, 120-amp-hr. battery in a very small machine, and all this winter I have had no trouble whatever in turning over the engine. I have had considerable starting trouble, but none so far as spinning the engine is concerned. I have a double-ignition system. The car has been frozen, and although the storage battery would turn the engine over at a rate of speed that ordinarily would start the engine, I could not get the proper carburetion or ignition to further aid the starting. I had an experience with some transmission grease which absolutely prevented the starting of the engine until the clutch was released. That may sound peculiar, but that particular grade of grease congealed, and prevented the starter from turning over the transmission—just the constantly meshed gears.

CHAIRMAN STRICKLAND:—I think it is in order to ask whether a transmission was locked onto the engines tested.

MR. OETTING:—In most cases, I believe, we did not have the transmission in. Some persons claim they can start better with the clutch in, and some with the clutch out. This does not affect starting as much as one might suppose.

A MEMBER:—My question may not be in order, but why use grease in the transmission?

CHAIRMAN STRICKLAND:—In winter, oil is like grease.

A MEMBER:—We have heard that it would be a good plan to have larger batteries; also that a majority of manufacturers object to the increased first cost of bat-

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teries. We have arranged for oversize automobile rims. Why would it not be a good plan to work out a battery carrier in the same way, so that the purchaser could buy an oversize battery, if he wished, and stand the extra expense?

A MEMBER:—That is all right, but there is no room for it under the front seat.

CHAIRMAN STRICKLAND:—If we could persuade the public and the sales force to let us place batteries on running-boards we could take care of that. The subject has been covered by reviews in automobile magazines, and it certainly would be desirable to have the battery in a more accessible position than it is on a large percentage of cars, and also to have room for a larger battery if an owner wishes to put it on.

MR. SCAIFE:—Is it not true that the Government is now specifying practically the largest six-volt battery that has ever been used?

Liberty Truck Batteries

MR. OETTING:—The battery used on the Liberty truck is of heavier construction. The capacity is 130 ampere-hours, and this battery, as tested by the Government, must stand one million vibrations in 30 hours. It is built to stand service, and does stand service.

CHAIRMAN STRICKLAND:—How many miles does such a test represent?

MR. OETTING:—I do not know any way of determining what one million bumps in 30 hours means in actual service. I feel sure that a battery to stand that service will be good for at least three or four years' service under actual running conditions.

A MEMBER:—What sort of vibration are they subjected to?

MR. OETTING:—The battery is mounted on a platform and dropped 5/16 in. 560 times per minute; that is one million bumps in 30 hours, and they have withstood a little over two million such vibrations.

A MEMBER:—The Government is specifying plates at least ¼ in. thick for the batteries used in naval work. They have tested all the thin plates up to ¼ in., and have decided that the life and reliability of a battery is dependent on the plate thickness, everything else being equal.

CHAIRMAN STRICKLAND:—Is there any difference in the life of a twelve and a six-volt battery?

MR. OETTING:—None whatever; the cells are all alike.

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The rate of current per plate is the same, and there is no difference in the gassing.

A MEMBER:—Why are batteries placed in the present position in cars, the lugs coming under the edge of the seat, where they cannot be reached with a wrench?

MR. OETTING:—The battery manufacturer does not design the cars, or put the accessories on. We have had great difficulty in persuading automobile builders to put the battery in some standard place. Cars come into our service stations that keep an expert service man busy for an hour changing the battery.

CHAIRMAN STRICKLAND:—Are the batteries that are placed under the skirting, those with a special long shape, growing in popularity?

MR. OETTING:—No; I think the square-type battery is the one that will come eventually if space can be found for it. It makes a better battery application, and permits better mounting of the battery in the box.

A MEMBER:—Is it not a fact that manufacturers generally ought to use larger batteries?

MR. OETTING:—That was one of my principal purposes in picturing these results, and was one of my conclusions. No doubt 75 per cent of the cars ought to have larger batteries. It would improve the starting and lengthen the life of the battery. The sizes recommended in this paper are probably not the last word on the subject; the table in the paper lists *minimum* sizes.

A MEMBER:—Why is a square preferable to the long assembly?

MR. COOK:—I do not know of any particular reason, except that it happens to be the type of battery most generally used, though I think it is better mechanically for the ordinary size battery. The case stands up better. From a battery standpoint, I think there is no preference.

A MEMBER:—The long assembly has one advantage on most cars in that it is much easier for the owner to reach and easier for the service station; they need not remove the cushion from the seat, and there is no risk of damaging upholstery.

Correct Location in Car

A MEMBER:—What is the correct location of a battery on the car?

MR. OETTING:—The place of least vibration is usually chosen, and I think the automobile engineers have placed the battery under the front floor-boards, or front seat, with the idea of gaining longer life there. In putting it

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under the seat of the car, they have chosen a fairly good place.

MR. SCAIFE:—Batteries under the front floor-boards are difficult to place in position and to protect from the heat of the engine in summer. I have known of the sealing compound on the battery melting owing to the heat from the engine and exhaust pipe.

A MEMBER:—It is impossible to protect this type of battery so that in washing the car the water will not get into it. Furthermore, it is impossible to locate the cells of a square battery so as to safeguard against short-circuiting between them. A long battery will obviate short-circuiting. With end-to-end assembly the positive and negative poles are farther apart. In fact, some 12-volt assemblies totally discharge themselves in two weeks without any connection other than leakage. That is of some importance. I think we have made tests on the ordinary square assembly and found it will discharge itself in 40 to 60 days under fairly bad conditions.

Lead versus Nickel-Alkaline Batteries

MR. MCGEORGE:—I hoped to hear something more of the low-temperature characteristics of the batteries themselves. I have had experience with electric batteries, but my experience with gasoline cars has been slight. The discussion regarding starting has been on the mechanical side of the motors themselves. In my earlier days I used nickel-alkaline batteries and had some bad trouble. Lead battery men told me it was because the battery would not stand the cold weather. At that time this battery was inclosed in a tight case to keep it warm, and I very carefully inclosed my batteries first in wood and then in steel to keep them warm. In talking to Mr. Edison he asked me about the construction of the battery box. He said: "All that I ask is that the draft shall be kept off."

Is it any worse for a nickel-alkaline battery to freeze than for a lead one? Tonight I hear for the first time an intimation from a lead battery maker that his battery efficiency falls off when the temperature goes down.

MR. OETTING:—It is entirely true that at 10 deg. the capacity is about 50 per cent. During some tests on lead and nickel-alkaline batteries at low temperatures the lead battery gave all that was expected at the 20-minute rate at 10 deg. while we could not hold the rate with the nickel-alkaline battery.

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A MEMBER:—I think some of the difficulty has been in not making exact comparisons. It is true that a lead battery will drop to approximately 50 per cent of its capacity, but the lead battery has lower internal resistance than the nickel-alkaline battery. The latter battery at low temperatures will not maintain its voltage at higher rates. I had the opportunity lately of testing some nickel-alkaline cells at low temperature when they gave only about 10 per cent of their normal capacity at a five or six-hour rate, while the lead battery gave 30 or 40 per cent of its capacity. In northern countries they take very great care, in operating nickel-alkaline batteries, to keep all compartments well inclosed, covered with felt and in some cases even keeping carbon lamps burning. I know of no cases where an electric vehicle using a lead battery has had to inclose or pad the battery to prevent it from getting cold.

A MEMBER:—If a battery is discharged and gives out 50 per cent of its normal capacity, will the rest of it be available if the battery is warmed?

MR. OETTING:—Yes.

A MEMBER:—When the ignition wires lead from a battery that is too small under cold-weather conditions, upon attempting to start the motor the voltage of the battery will drop so low that sufficient ignition cannot be obtained. I have seen attempts made for ten minutes to get a car started whereas the battery would start the engine immediately if the crank were used, allowing the battery to furnish only the ignition. This shows that instead of trying to exhaust the battery one could start under bad conditions by using the battery for ignition only.

A MEMBER:—Is not the failure of the ignition due to the fact that at the time the spark takes place the greatest draw on the battery is by the starting motor.

MR. OETTING:—When using the proper size of battery for starting, the voltage will not go down so as to make the ignition faulty.

A MEMBER:—It was mentioned that these starters could not be designed to be more efficient at the low speeds; now, what change could be made in the starter to make it more efficient at high speeds?

MR. OETTING:—It is a matter of proportion of iron and copper in the motor. The point I made was that a design for cold weather will work at normal temperatures, 70 and 80 degrees.

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A MEMBER:—Could that not be attained at less expense by having greater reduction?

MR. OETTING:—Yes, that was one of my conclusions. I think in a double-reduction there will be a higher break-away torque. It is not the spinning power but the high break-away torque that starts the engine. During a test made not long ago on one of the largest engines thus far adapted to electric-starter service we used a double-reduction on the flywheel drive. At normal temperature the motor turned over at 90 r.p.m.; in cold weather at 15 deg. the engine turned over at 75 r.p.m. The double reduction was 17 to 1 and the engine four-cylinder with 5-in. bore and 7½-in. stroke. The armature of a starting motor on a starter of this kind must be designed to have a high acceleration so that there will be no backlash on the flywheel.

PRESENT REQUIREMENTS OF THE AUTOMOBILE USER

BY J EDWARD SCHIPPER*

The war is like a gigantic metal whirlpool into which metals are drawn from all over the world to find their way into the guns and into the thousands of other war instruments which are necessary in the carrying out of a modern battle.

Many of these metals and much of the other materials used in the construction of automobiles are used at the front; other metals and materials used in the automobile are needed in the factories making munitions for the front; hence the question of materials becomes one of prime importance in the automobile of to-day.

This may seem like a matter entirely for the car manufacturer, and yet it has as much to do with the customer as with the maker of a car. In spite of the fact that cars are increasing in price almost from month to month, the demands of the user for performance are also increasing. While the average price of automobiles has gone up several hundred dollars, if we leave out of consideration for the moment the very lowest priced cars, the average price that the user wants to pay has not

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gone up nearly as much as the price, and therefore we see performance, which was formerly expected only in the higher priced cars, demanded as a matter of course from cars which at the present time do not cost nearly as much.

No one knows better than the automobile manufacturer that the car buyer wants performance. The carbureter manufacturer knows that the public is demanding an instrument capable of throttling to speeds hardly in excess of 2 m.p.h. and upon application of the throttle of picking up rapidly and surely to speeds in excess of 50 m.p.h. on fuels which are continually going down in quality and volatility.

What does the average user demand? He wants a car that does not cost him too much, in the first place; he wants one that will be in the service station for the minimum amount of time. It is a matter of pride with every user that his car is dependable, and it is a matter of necessity with most users that the car be able to perform more than 300 days a year without a falter. If we took an average of all the users in the country, we would probably find that there is a demand for 15 miles to the gallon of gasoline; there is a demand for quick acceleration, good hill climbing, easy gear shifting, comfort in the seats and springs, accessibility for small adjustments and repairs, a minimum or rather an absence of rattle. When it comes to tires, anything short of 8000 miles is always occasion for a fit of peevishness.

Size and Acceleration Demanded

The American public demands the appearance of great size in its cars. The largest car selling for a given price will sell best simply owing to its size. Coupled with this demand for size, the American public also demands acceleration. Size means weight and weight certainly operates against acceleration to such an extent that the torque requirements of the engine must be increased greatly to give the pickup desired with the large car.

In Europe there are cars which, by a scientific use of the wheelbase, give all the necessary room at a greatly reduced weight. These can supply all the acceleration demanded with an engine of greatly decreased horsepower. The fact that the car weight is less allows reduction of the engine bearings and all other parts, and at the same time factors of safety are largely those used in average American practice, giving greater endurance and a smaller service factor than can be expected from

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the large, heavy, high-power design which engineers have been forced into much against their will.

War time is a time for economy and efficiency; therefore, let us appeal for a car design which is efficient and economical. It is exactly what the public wants, but owing to the education given them through unhappily worded advertisements and by other means, the public believes that its demands are met in proportion with the size of the car. Tire economy and miles per gallon of gasoline and other desirable requirements have been sacrificed because it is believed that acceleration and comfort in riding can be secured only with the car of long wheelbase. As a matter of fact, careful design, coupled with compactness in engine units, will allow several inches to be taken from the wheelbase of the car without any sacrifice of room, without great loss in riding comfort, and at the same time with a gain in materials, a saving in weight, a saving in tires, a saving in gasoline, a saving in space required for parking, and a saving in every direction which can be thought of. It costs less to build a shorter car, and this money could either be taken from the price of the car, which would certainly be another factor in what the user is demanding, or it could be used in refining the design, probably with the result of even greater lightness, better upholstery, better painting, and better trimming; all tending to make the result a high-class car throughout.

Trend Toward Smaller Cars

High-priced cars in this country have nearly always been big cars, owing to the necessity of catering to the taste for size, and yet the very trend of public demand shows that exactly what it wants can be met in a medium-priced small-sized car which incorporates all the beauty of design to be found in the highest priced products, and all the excellent materials used in cars which today are made with a long wheelbase.

Engine size has nothing to do with performance. Torque-weight ratio is the governing factor in acceleration and hill climbing, the two greatest demands made by car buyers in this country. The demand for reserve torque will probably persist for years to come, and the value of horsepower is something which has been greatly exaggerated in the literature of engine and car manufacturers. The words "high power" are fascinating to the layman, and he has been led to believe that if he has

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a powerplant capable of a high maximum output under his hood, he has an engine which will enable him to run away from others not capable of developing such high output. This, of course, in the face of close analysis is untrue, because the high horsepower may not be reached until the number of revolutions per minute and the car speed become prohibitive under ordinary gear ratio conditions.

All the performance desired in a five-passenger automobile can be secured with an engine having under 200-cu. in. piston displacement. All the acceleration and hill-climbing ability could be developed in such a car. The ratio of engine torque to weight could be made very high at the speed most desired. If an engineer were told to get out a five-passenger car incorporating all the performance demanded, together with an economic use of materials and an economic maintenance factor, he would err in going much above the 200-cu. in. limit. The idea that a perfect-riding car is one of great weight should be abandoned.

Pounds per Passenger Too High

Weight of car per pound of passenger weight is too high in America. The tendency toward long wheelbase and big engines is responsible for this. A five-passenger car should not weigh more than 3 lb. per lb. of passenger weight if it is to be considered a truly economical design. With a designed load of 750 lb., a 2250-lb. car would be about right.

This presents the picture from one side. We all know what the public demands because we know exactly what we like ourselves in a car. We like to step into a roomy, comfortable front compartment, in which we do not have to squeeze ourselves behind the steering-wheel. We are always pleased when our first glance shows that the control members can be easily seen and reached. We want the engine to start almost on the first touch of the starting button; we like the starting pedal to engage with little effort, with the engine running smoothly and quietly; we want a clutch pedal which is depressed by a very light touch of the foot and which when engaging gives a smooth pick-up without unnecessary slipping and excessive grabbing. In shifting gears, the clutch should release instantaneously and positively, so that there is no drag in the gears, and the gear lever itself should throw into mesh with minimum movement and minimum reach to the shifter lever. In picking up, the

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operator should be able to depress the throttle quickly without creating undesirable conditions of knocking, choking, spitting back or laboring of the engine. In other words, carburetion, gear reduction, and engine capacity and torque output should be so correlated that a smooth accelerative ability results. Here, of course, lightness again enters.

Cold-Starting Problem

To the list of requirements already given must be added that of starting rapidly on a cold morning. This is a most pertinent point at the present time.

All the hot-spot intakes and exhaust jacketed carbureters in the world will not do one thing toward making the car easier to start. The carbureter manufacturers have provided us with very efficient chokes, yet even at the speed attained by the average starting motor on a cold morning, these chokes shoot into the cylinders a considerable supply of raw gasoline. Sometimes part of this raw liquid is ignited during the first three or four minutes of cranking, but often the motorist helps by distributing ½ pint more between the four, six, eight, or twelve cylinders. This joins the supply which the efficient choke has put into the combustion space, and ignition starts; only, however, after a liberal supply of the raw fuel has leaked its way past the piston rings and gone down into the crankcase.

If the car starts in 30 seconds the user is fairly well satisfied. He does not think, however, of all the gasoline that can be drawn into the cylinders and leak past the piston rings in the lapse of 30 seconds; he does not consider the fact that his lubricating film has been destroyed to a large extent by this gasoline; and he does not stop to consider how much the supply of lubricant in the crankcase has been deteriorated owing to dilution of the oil by the heavier constituents of the gasoline.

This situation is probably the most serious one which confronts the designer to-day. The automobile user is demanding service, and he will not get good service from an engine which is being treated in this manner. The first few minutes of running with an engine in this condition do considerable damage. The problem must be met quickly.

In these days when high prices of labor and material are forcing the price of cars upward, something must be done to meet the cold-starting situation. A heating coil in the float chamber or other gasoline chamber, a water-

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heating system connected with the radiator, or some other method which is equally effective, must protect the owner from the consequences of crankcase dilution. The average owner knows that it requires a pint or more of gasoline to warm up his engine on a cold morning. He knows that he is not burning this amount, and all of it is not going out at the exhaust; the remainder forces its way into the crankcase. Cars fitted to combat this trouble will go a long way toward meeting one of the biggest demands made by car users.

Preheating the Air

The heated intake has been studied so carefully by all manufacturers that it is useless to dwell upon it. However, there are a great many manifolds which function on the side of too much heat rather than too little. It is practically impossible to meet the extreme ranges of temperature conditions—100 deg. or more in most of the cities in our territorial belt—without some sort of adjustment. The air is always preheated to some extent and probably the means of adjustment to meet the outside temperature conditions is most rapidly effected by varying the amount of heat which is imparted to the initial air before it enters the carbureter. Too much heat makes the engine unsatisfactory, due, of course, to the reduction in volumetric efficiency, but the proper amount of heat, in spite of the expansion of the intake gases, will often result in an improved torque curve.

Economy and Appearance Demanded

It is a noticeable fact that this year's car users are demanding gasoline economy more than ever, and there is vigorous activity on the part of carbureter manufacturers in supplying this demand.

The carbureter, however, is not the only factor in economy. Lightness is a paramount necessity. Lightness does not mean alone reduction of weight by scientific design of parts; it is possible to decrease the weight far more by a scientific use of wheelbase length. In this country we have been wasteful of wheelbase. We have secured a minimum of passenger space with a maximum length of car. It has been possible in Europe, due to the use of shorter engines, shorter hoods, and the more scientific location of parts, to get much more room on a given length of car, with the result that the car weight per passenger has been very much reduced.

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Scientific designing from one end to the other would make possible a great economy in materials with very little sacrifice in performance; in fact, there need be no sacrifice but an actual gain in the way the car operates.

The rear axle gears and in fact the entire transmission mechanism should be quiet and sturdy, the propeller-shaft should not whip, and none of the parts beneath the chassis should be so filled with grease cups that the user is fortunate, indeed, if he finds 25 per cent of them on his first greasy search.

The appearance of the car should please the eye. Exterior appearance is a matter of choice, and as taste changes from season to season, it is imposible to lay down any rule. Present-day style calls for long, straight lines without sharp breaks from radiator cap to the end of the tonneau. Another year this may be different. Nevertheless, the car should conform to the dictates of the moment. The upholstery should be good, and here it may be said that some of the latest of imitation leathers are rendering excellent service and are doing better in fact than poorer grades of genuine leather.

The top should be designed to drop quickly into place and the side curtains should permit of ready view all around the car. Some side curtains with their small openings shut off the view of the driver, and are a positive menace in city driving. The windshield should be drip-proof and have the ventilating feature. The engineer must supply by scientific design what the public demands in spite of the fact that he is now under limitations of material which did not hinder him a year ago.

THE DISCUSSION

A MEMBER:—What has Mr. Schipper found regarding the sentiment for permanent tops or enclosed bodies—are they increasing or decreasing?

Mr. SCHIPPER:—I believe the use of the permanent top is increasing, and by that I mean a top that *is* permanent. The winter top, or the one that is put on in place of the summer top, is disappearing. I think the convertible car, in which the sides but not the pillars are removable, is increasing, and construction is materially improving on that type. The pillars are being made more rigid and they are fastened in place more substantially. The car will remain tight for a longer time than used to be the case. The tendency to rattle was so great in these convertible sedan types that it threatened

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to kill the design, but improvements have come along fast and I believe are on the increase.

J. A. WILLIAMS:—I do not believe that they have yet found an all-weather car. As yet there are no means of supporting a heavy, bulky top with the satisfaction of a solid brougham sedan that was first brought on the market, but I believe the ultimate car will answer all those purposes. The doors will probably have flappers, but I cannot believe that the ultimate car will have detachable pillars, because they are inconvenient to handle.

W. R. STRICKLAND:—The feature of removable pillars is handled differently by different owners. In some parts of the country they take the pillars out and have no occasion to put them in quickly. In other places where the weather is unsettled the removable pillar feature has helped the salesman to satisfy the buyer. I believe the permanent pillar is an improvement from a body standpoint.

A MEMBER:—It has always been my idea that the ultimate enclosed car will have no pillar on it. It seems to me a car can be constructed on which the glass in the front door will fold in and disappear; and the glass in the quarters between the rear door and the front door will fold in with the rear door, and the rear quarter glass will fold in and around and disappear behind the seat. I am trying to work out that proposition at the present time. I really think removable pillars are very objectionable. There is rattle and strain on all pillars in spite of the precautions we take in tightening them and putting in bumpers. The one-man top has not been a success, and the ultimate stationary top will be made with no pillars but with glass that will disappear.

Body Construction

Mr. WILLIAMS:—Speaking of the ultimate car, the percentage of sales is very small on enclosed bodies. Hardly 10 per cent of the cars are driven to-day with the top down. Top materials cannot be folded up and down continually without cracking and it is evident that tops will in the end be permanent.

A MEMBER:—The ultimate car will not have a folding top. Buyers are going to demand better protection from the weather. The body designs have been changed from year to year; the engineers keep improving the chassis and still we go on with the same old top. If we can design a neat-appearing top that does not have to

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come down, I think it will appeal to the average car buyer.

Mr. STRICKLAND:—In Mr. Schipper's paper a good deal was said about economy. Folding doors and windows increase the weight tremendously. Most of the features mentioned increase the wheelbase. We are trying to keep the weight down and keep the economy good. Dealers, I think, are coming to the conclusion that these heavy cars will decrease in popularity owing to their weight. People do not want to go touring with such heavy cars. It means too much wear and tear on the engine and tires and too much gasoline.

Mr. WILLIAMS:—That is an excellent point, but the most obvious economy has been overlooked and that is the initial cost as compared with that of the present touring car.

A MEMBER:—Mr. Schipper said that carbureter makers have endeavored to increase the velocity in their carbureters, thereby raising the volumetric efficiency at lower speeds, which is quite right. From my experience one is not satisfied with quicker acceleration, good torque and good performance of car at low speeds. In addition very good high-speed performance is desired. This problem has not been solved yet, and becomes harder every day as the fuel we are obliged to use becomes poorer.

Multi-Stage Carbureters

Mr. STRICKLAND:—Do you suggest the two, three and four-stage carbureters?

A MEMBER:—The two, three or four-stage carbureter will not handle the proposition because the velocity will not govern the volumetric efficiency. With a manifold in keeping with the two-stage carbureter, if it is of such small size as to correspond properly with the first stage, the volumetric efficiency of the same engine in the second stage will be very low because the manifold velocity is too low. I have made some experiments this last year, and have found that maximum torque can be obtained on an engine with very large valves with the first stage carbureter, and maximum horsepower can be obtained.

Manifolding will not take care of the gasoline we have to struggle with today. I do not agree that a thorough atomization is of any value. If heat is not applied to the products of atomization it does no good at all with the present gasoline. The opinion of engineers as to heat applied at a certain spot is varied. We have to

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figure a theoretical loss of volumetric efficiency as soon as we apply heat to the gasoline, but this small amount can be neglected if we will figure how much actual thermal efficiency we gain by having to deal with a gas instead of with a fluid.

Mr. SCHIPPER:—I saw lately some extensive experiments carried out by the Holley Carbureter Co. on the Ford tractor. It was found possible to vaporize kerosene, get it in the form of gas, and then carry that gas even with a drop in temperature, through several inches of the manifolding without recondensation. If that can be done with kerosene it can be done with gasoline. I believe Mr. Horning once mentioned the same thing in his experience. If the kerosene is put in the vaporous condition it can be carried over the long distance without condensation taking place, as verified on the Ford tractor. Their first design had the shortest possible intake, because they believed that if they lengthened the intake they would get condensation, but they found that the longer the intake, without limits, the higher the performance on the block will be.

Referring to engine performance and economy, one of the reasons our engines are not more economical is that the piston displacement is too high. In a five-passenger car this should not go above 200 cu. in., and it is my firm belief that the tendency is toward decreasing piston displacement. On the Indianapolis Speedway a car of that order gave a performance that startled the public. That was with a little Peugeot, and if it can be done in racing cars it surely can be done in passenger cars.

Kerosene as an Anti-Freeze

I want to mention the using of kerosene for an anti-freeze solution. Several factories are so using it with a pump cooling system and when the system is kept full, it works out all right. With a tubular radiator it works out very well, that is, on the cars of the Ford and Dodge type, but on cars with the thermo-syphon system and cellular radiator such as the Hupmobile, the kerosene does not seem to work well; the explanation given is that the pressure due to the thermo-syphon is too low. The prices of alcohol and glycerin are so high now that I really think the engineers in the industry ought to inform car owners immediately. A good many have gone to calcium chloride sold under different names and are troubled with short-circuited ignition systems.

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A MEMBER:—Four years ago we had a Chalmers 36 and we used kerosene all one winter. That had a pump circulation and we had very good success with it.

Mr. SCHIPPER:—Testers around the Chalmers factory are using it in factory cars.

Mr. STRICKLAND:—I think the greatest objection would be the odor of the kerosene. I never thought of using it before, but in really cold weather it would save considerable money.

A MEMBER:—I have never used kerosene, but should imagine that one of the chief difficulties would be in keeping it thoroughly mixed with water.

Mr. SCHIPPER:—Do not use any water—use all kerosene. The specific heat of kerosene is, I think, about 0.7 that of water, so that the engine will be kept running at a high temperature with the same amount of radiator service, and for that reason the temperature is going to be higher. For every pound of kerosene circulated, instead of carrying off a similar amount of heat from the cylinder walls, 0.7 of the amount that would be carried off by water is removed.

A MEMBER:—A compound that is soluble in the kerosene might be mixed with it, using enough to make it the same density as the water.

A MEMBER:—A friend who had filled his radiator with kerosene said he had run his car up to 40 miles an hour and had no trouble with kerosene except that it melted out the rubber hose. If it got very hot kerosene might cause trouble with vaporization.

Mr. SCHIPPER:—We can count on losing a set of hose a season with kerosene, nevertheless the money saved would replace the hose many times. It would not do to use kerosene at temperatures over 20 deg. fahr.

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DETROIT SECTION PAPER

RECENT TREND IN AUTOMOBILE DESIGN

By J Edward Schipper*

War demands a speeding up of movement. The methods of the lax times of peace do not suffice for the quick results demanded in war time. Under the demands for the movement of unprecedented quantities of materials and manufactured products, not to mention the greatly increased passenger traffic of a military and commercial nature, the facilities of the nation's railroads and travel arteries have been strained to the breaking point. With the railroads of the country taxed to their utmost limit, automotive transportation must come to the rescue.

A century ago ideas of distance were gaged by the abilities of the horse. Although over one hundred million acres of land are cultivated annually in this country to support the horse, nevertheless the days of thinking in terms of horse transportation are done. Quick, certain and, above all, efficient transportation is the greatest demand of the age. Let us see how the automotive industry fits in.

Change From Peace to War

We have been awakened from our gilded dream to find that our Chinese wall of self-sufficiency has crumbled, largely because transportaion has advanced to such an extent that countries, which in the days of "no entangling alliances" were remote, have become our neighbors. This great change has been accompanied by a revulsion of feeling in this country which a few years ago would have been considered impossible.

Our national fault has been extravagance. The great forests which once covered the country with timber have been cut away so that today lumber is cheaper in the oldest countries of the world than it is here. Our minerals, the game from the fields and forests, all the gifts that nature bestowed on this country have been used with an unsparing hand, and naturally along with this the habits of our people have been molded until we have become known as the most wasteful Nation on the globe. It is but natural that this molding of our nature should express itself in the tools we use. It is shown

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quite clearly in our automobiles. The American people have been buying cars by size and weight. Two cars, each costing the same, but one larger and more gaudy than the other, if placed side by side and offered for sale would not compete, as the larger one would command nearly all the attention in spite of the fact that the other might be a better piece of engineering from end to end. But a change has come. A great wave of economy has swept over the country.

Recent Changes in Public Attitude

We have suddenly awakened to the fact that the automobile of today is inefficient and uneconomical, and, at the same time, the people have been aroused by the exigencies of war to the fact that their daily habits *must* be more efficient and economical.

The result is natural. People now want the most for their money. When they buy transportation they want the most transportation for the money expended at the time of purchase as well as afterward in maintenance and operation.

The meaning of *efficiency* is clear to any engineer. It is simply output in terms of input. If a machine is 80 per cent efficient, its output is 80 per cent of its input, whether measured in terms of heat, distance, dollars or any other of our standards of measurement. When we say automobiles of today are inefficient, and are not returning to the owners proper values in transportation for the money expended, we simply mean that what the owner gets back in transportation is not enough in proportion to what he puts in.

Take, for instance, the man who owns a car weighing 2½ tons, who drives downtown to his office, a distance of 5 miles, and in the evening returns, another 5 miles. If a friend were to offer to take him to his office and back for a dollar a day he would probably laugh at him, yet, in all probability, this is what it is costing him. The upkeep of such a car, even without including the high depreciation of a car of this weight, will easily amount to 10 cents a mile.

The car weighs too much for the passenger load. The man weighs probably 150 lb. The car weighs 5000 lb., that is 33 lb. for every pound of passenger weight. It is true that the car is heavier than the average, but this example is taken merely to illustrate the point. With five passengers in the car the weight would be over 6 lb. per passenger, and even this is too much.

Considered from the transportation efficiency stand-

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point, there are two sides to the ledger, one showing what we put in and the other what we take out. We put in money in terms of fuel, time, weight of vehicle, labor (if a driver is employed), road or track maintenance and vehicle maintenance. We take out just two things, passenger-miles and comfort. Comfort costs extra on an automobile as it does on a Pullman car. In a commercial vehicle we take out ton-miles alone.

Cost Factors of Operation

Most of the cost factors are in the control of the engineer; indeed, most of them are simply weight factors. Considered one by one, they are:

Fuel Consumption—Primarily affected by weight and to a moderate extent in average practice by design.

Time—Controlled by law and traffic conditions.

Weight of Vehicle—The crucial factor involving displacement and weight of powerplant, length of wheelbase, physical qualities of materials, vehicle characteristics and purposes.

Road or Track Maintenance—Covered by state tax and in many states a function of weight and horsepower.

Vehicle Maintenance—Factor of weight, materials, design, handling, road conditions and a number of other variables.

In the past all of these items have been high and we must make rapid changes to reduce them if we wish to fit in with the desires of the public.

No Real War-Time Cars

The new cars at the shows are hardly war-time cars. They have been under test out on the road for so long that they antedate the economy wave which has swept over the country. They serve to show only that the tendency was in the direction of this new demand. In a great many ways this demand is not even known to the people themselves. The attitude of the people at the shows was to a large extent noticeable. They were searching for something they did not find and so they took the nearest thing to it, or nothing.

These people were looking for a car in which the weight per passenger was at the lowest limit. In other words, a car that utilized every inch of wheelbase to the greatest possible efficiency, that was not of a displacement higher than necessary to carry the load, that had materials which returned the greatest amount of

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service because of their light weight and scientific disposition.

No longer were they looking for the biggest car that they could buy for a certain sum of money. The American public has had a psychological shock from which it has not recovered. It behooves the automobile industry to design to suit this new state of mind.

At this time of the year when all the new cars have made their appearance and the designs for the following season have been disclosed, we are naturally interested in noting the trend of development. This year, for the reasons pointed out, the trend cannot possibly teach us as much as in the past because the designs in which they are included cannot possibly have taken into consideration the national psychological change. Nevertheless, since we were beginning to move in the right direction it is important to study these developments to see what should be accelerated in order to meet the new conditions and new standards of conservation and economy.

At the shows there were but five new chassis incorporating new powerplants and redesigned to a sufficient extent to classify them as being relatively new. During the year two additional chassis which can be classified as really new were also brought out. This gives a total of seven entirely new chassis which were not on the market a year ago. All of these have new powerplants and are the latest types under normal development. It is interesting to note that of the seven, five are four-cylinder and five have overhead valves.

Probably one of the results of more carefully shaped combustion chambers, is that mean effective pressures have been increased, combined with a tendency toward larger valves, more scientific cam contours, better designed intake ports and galleries, and superior carburetion owing to the hot-spot manifold and carbureter mixing chambers giving better velocities.

Shorter Wheelbase Probable

If automobile factories carry out designs suggested by the changed attitude of automobile buyers, the changes to be made during this season will be more radical than anything in the last decade. There has been no incentive for the engineering department to make every inch of wheelbase return an inch of body space or other concrete qualities, as riding comfort. As a matter of fact we have had more than enough wheelbase. At the present cost of materials and with the present objection of the

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public to carrying around any more useless car weight than is absolutely necessary, manufacturers are not going to be so wasteful of car length.

What can be done in this respect is hardly appreciated. For example, one company which brought out an entirely new car this year shortened the wheelbase 7 in., and, after rearranging the layout, had as much room in the tonneau as in the previous model, and 2 in. more in the front compartment. This was not done with the idea of cutting wheelbase length to the limit, but rather was more or less incidental to a redesign. In addition to more passenger space and a smaller engine, a gain of 50 per cent in economy of fuel, a large gain in weight, and a better performing car were secured. This better performance is in acceleration, low-speed torque, and speed range on high gear.

In an analysis of this kind it is only fair to remember that it has been impossible to realize many of the anticipations of a year ago owing to the fact that the minds of engineers who would ordinarily be on development work, have been employed on the vast task of designing and making ready for government production necessities. Scarcity of materials has likewise retarded development this year. Without doubt, aluminum would have been employed to a greater extent than ever before, but the fact that it remains scarce and expensive has prevented its use on any but high-priced products.

Average Engine Design

If we were to take all the American passenger cars on the market and combine them into a general average,

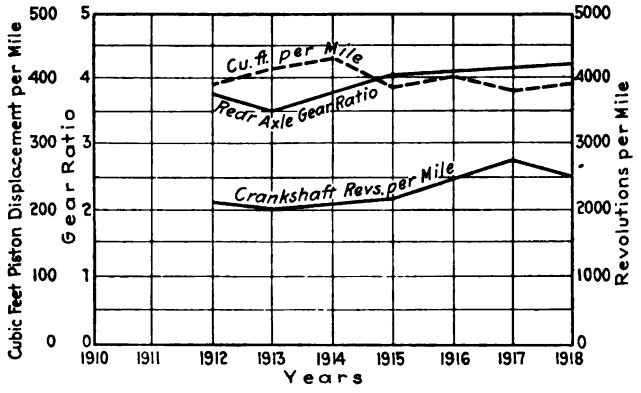

FIG. 1—YEARLY TRENDS OF AMERICAN PASSENGER CAR DESIGN IN PISTON DISPLACEMENT AND CRANKSHAFT R.P.M. PER MILE AND GEAR RATIOS

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the cylinder size would be 3½ by 5 in., and the average displacement, based on the relative percentages of numbers of cylinders, would be 269 cu. in. From the standpoint of engines of different numbers of cylinders the following averages obtain:

Number of Cylinders	Bore In.	Stroke In.	Piston Displacement Cu. In.
4	3.6	4.95	221
6	3.46	5.00	282
8	3.13	4.59	282
12	2.87	4.96	386

Contrary to what has been outlined as the probable future in the development of passenger-car engines, the average engine is larger today than it was a year ago. The piston displacement is now 269 cu. in., as compared with 222 formerly. The gear ratio has been altered from 4.15 to 4.22, yet the increase in average tire size from 31 to 34 in. has prevented a large increase in the displacement per mile on high gear. A year ago this averaged 384 cu. ft.; it is now 388 cu. ft. Thus, although actual engine displacement has increased, car ability, as measured by displacement per unit of distance, has not increased.

Overhead Valves Increase

A census taken before the shows indicated an increase in overhead valve construction of 2.2 per cent with the corresponding reduction in L-head engines. The additions to the list as brought out by new cars at New York and Chicago bring the increase in overhead valve engines to about 3 per cent. This percentage is rather lower than it ought to be because the assembled cars practically all use L-head engines, due to the fact that the largest builders of stock powerplants confine their efforts to the L-head type. As each selection of an engine represents what might be called an engineering ballot in favor of that type, it is fair enough to take the chassis and obtain the average as given.

The conclusion must be reached that an unqualified indorsement has been placed by the industry upon the overhead engine. The fundamental reason behind this development is naturally the effort to secure the greatest possible output from any given displacement. High volumetric efficiency, with lowest possible losses through wall cooling, would naturally tend to favor constructions in

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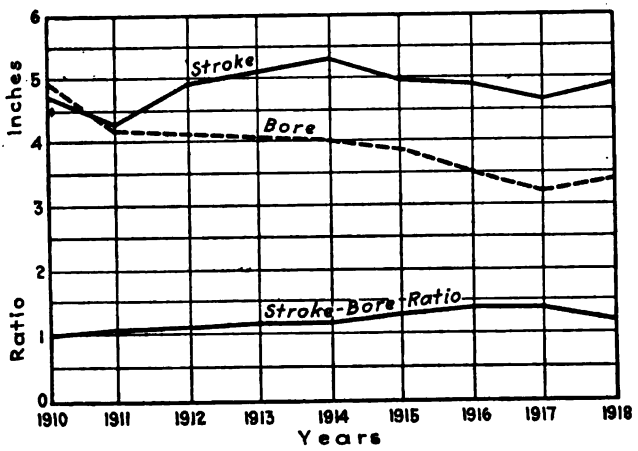

FIG. 2—AVERAGE CYLINDER SIZES DURING THE PAST EIGHT YEARS

which it is possible to use the most direct entrance into the combustion chamber and at the same time secure the most advantageous combustion chamber shape.

All the new engines brought out during the year have detachable cylinder heads; makers who have not heretofore employed this feature have now adopted it. Two Detroit makers, one manufacturing a high-grade twelve, and the other a high-grade eight, have changed over to the detachable head, showing that its practicability is recognized for the V-type engine as well as for the

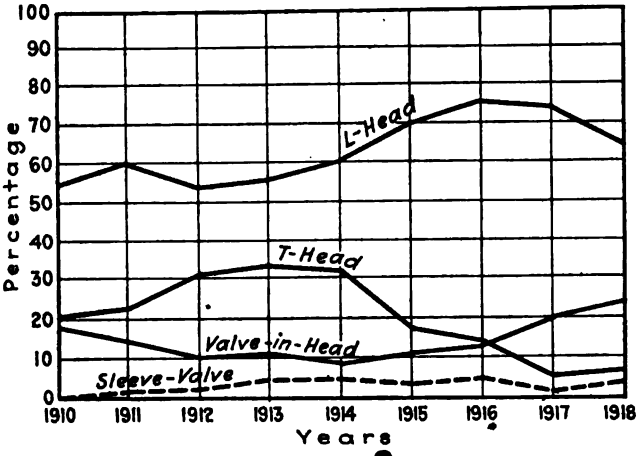

FIG. 3—THE TREND OF VALVE CONSTRUCTION ON AMERICAN PASSENGER CARS

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vertical. Early difficulties have been solved, and they have been found to be due mainly to two causes: either insufficient metal around the studs or poor distribution of water space around the head.

Fuel Question and Design

For three years the fuel question has exerted a highly important influence on engine design. This year there has been a crop of the so-called hot-spot manifolds. There is no doubt that the use of these has resulted in a material reduction in the warming-up period. This is of considerable importance, as with the highly efficient choking devices employed on up-to-date carbureters a stream of raw gasoline is drawn into the cylinder during the

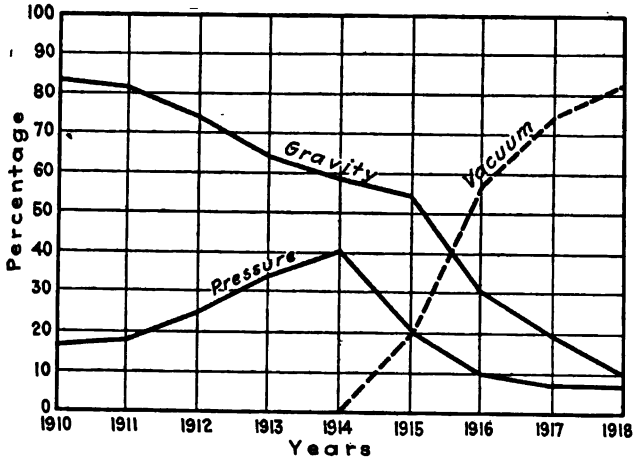

FIG. 4—CURVES SHOWING THE TREND IN APPLICATION OF GRAVITY, PRESSURE AND VACUUM GASOLINE FEED SYSTEMS

cranking period, and naturally some of this leaks past the piston rings, with the result that considerable crankcase dilution occurs, destroying the lubricating qualities of the oil.

Cold-Weather Starting

Cold-weather starting is engaging the attention of automotive engineers. During the year many suggested means have been offered for solving the difficulty. Probably the most promising of these is the method by which a small electric coil is employed to heat a small quantity

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of gasoline, which is thus evaporated, throwing into the cylinder a drier product which facilitates combustion and at the same time is more free from the evils of crankcase dilution. In one of these primers the electric coil is in the float bowl of the carbureter, and in another a cylindrical well is employed of slightly larger diameter than the pencil-shaped coil.

Single ignition has been continued, and has been improved in detail. The battery, breaker mechanisms and coil have all been improved. One company has developed a battery in which rubber separators are used with threaded wicks, which carry the active material in the desired manner. Breaker boxes and coils operate faster than ever, this being necessary to take care of the increased crankshaft speeds at which engines operate.

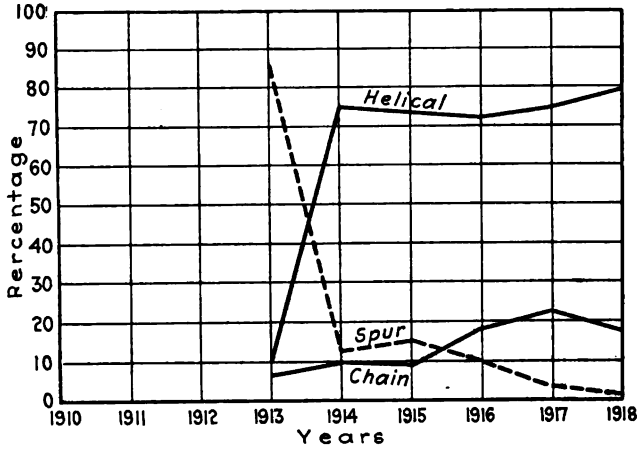

FIG. 5—TRENDS IN THE DRIVING GEARS OF AUTOMOBILE ACCESSORIES

Manufacturers have difficulty in keeping up the quality of breaker points, owing to the scarcity of platinum and tungsten.

Engine Balancing

Engine balancing has been studied more carefully than ever, and counterweighted four-cylinder engines are now more numerous. Inherent balancing, in which the rotary balance of the shaft is taken care of in the design, is also meeting favor. It is interesting to note that the use of pressure feed for the main bearings is increasing. While the percentage does not show up very high, owing to

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the fact that there have been some new models added, which neutralize the figure, the proportion of engines with drilled crankshafts is increasing. In practically all cars selling at prices which permit drilling the shaft this method is being used.

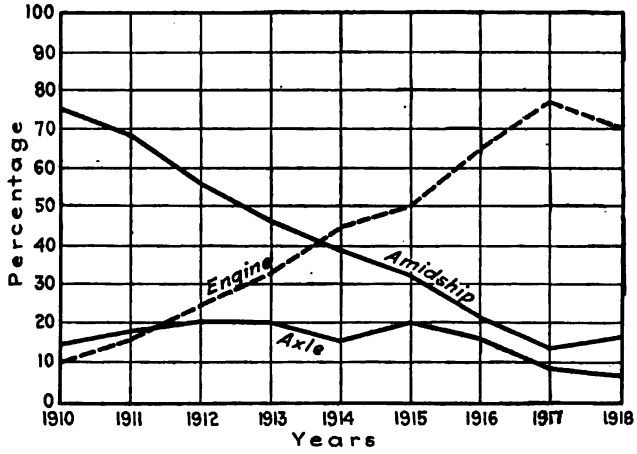

FIG. 6—TRENDS IN TRANSMISSION LOCATION

It will be noted that the average wheelbase has increased by 7 in. It is safe to assume that this will not be the case another year. The fundamental reasons noted at the beginning of this paper preclude the possibility of again adding weight unless there is a very good reason for doing so.

CHASSIS WEIGHT

As a matter of fact, the 1917 chassis has not followed the curve of weight reduction which past trends would indicate. The abandonment of alloy steel wherever it is possible to get a reasonably satisfactory substitute in carbon steel may have something to do with this. Another reason for increased weight is in the deeper frame sections which are now employed, although this depth is necessitated by the longer wheelbase. The increased depth has been accompanied by a tendency toward longer springs of the half-elliptic type. The latter form of suspension in combination with the Hotchkiss and other drives is growing in popularity. There remain only three cars which use full elliptic springs. Three manufacturers also continue to use platform springs, there having been

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a notable conversion in this respect of one of the companies making twelve-cylinder cars. Hotchkiss drive is used on 53 per cent of present-day American chassis.

Success of Light Clutches

Clutch adjustments continue to decrease. The success of the light types now on the market has been proved.

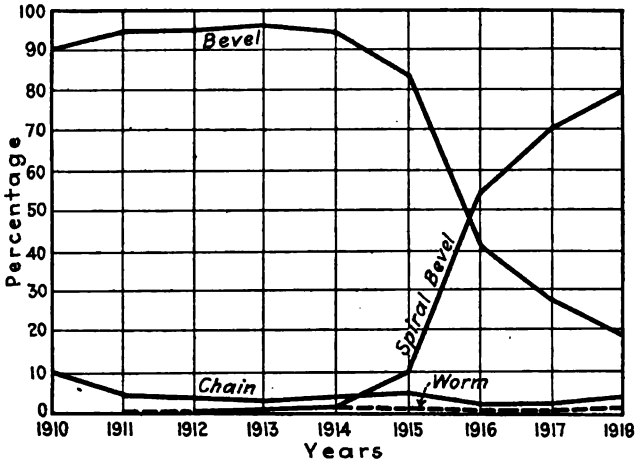

FIG. 7—TREND OF FINAL DRIVE DESIGN SHOWING POPULARITY OF SPIRAL BEVEL DRIVE

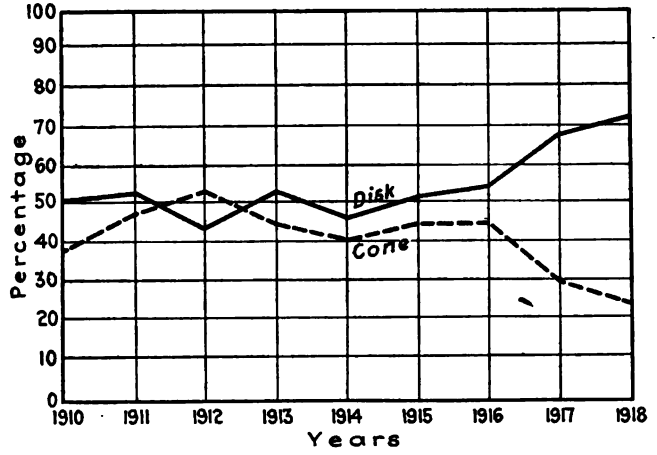

FIG. 8—DISK AND CONE CLUTCH TRENDS

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Securing the lightest and most efficient clutch to transmit a given torque has been a problem studied very carefully during the past few years, and at present the disengaging method has become very light and quick. The fact that on some cars with powerful clutch springs the clutch

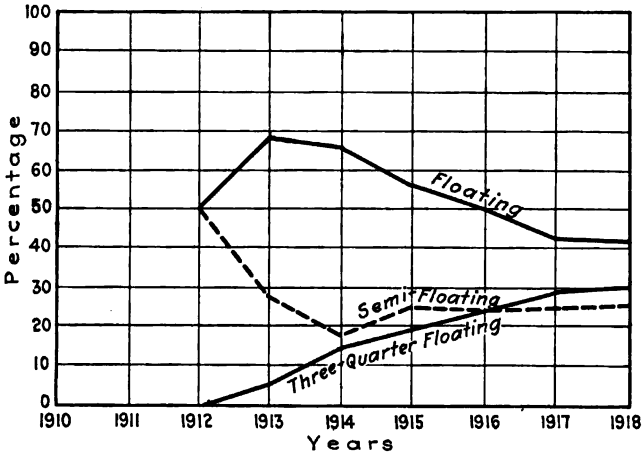

FIG 9—TRENDS IN REAR AXLE DESIGN

pedal can be disengaged with the hand indicates how the travel of the engaging members has been shortened. The use of fewer disks has contributed to the shortness of travel, and is one of the factors leading toward lighter clutches.

Materials for Chassis Fittings

In chassis fitting a marked trend toward the more extended use of forgings for brackets and axle parts has been somewhat checked by the abnormally high prices asked for forgings at the present time. The result has been a temporary tendency toward increased use of malleable-iron parts. It is not probable, however, that this tendency is any indication of a permanent condition. Manufacturers have been forced, in some instances, to change from forgings to malleables, and in a few instances from malleable to steel castings, simply because of the difficulty in obtaining delivery of certain parts.

Further than this, in view of the present prices of forgings, malleable iron is being used for some parts where the size is a matter of convenience rather than of

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structural strength. In a great many trucks which ordinarily would employ forgings, practically all of the frame brackets, spring hangers and engine brackets are of malleable iron. All parts which are in tension, and particularly those which are under shock, such as rod ends, are

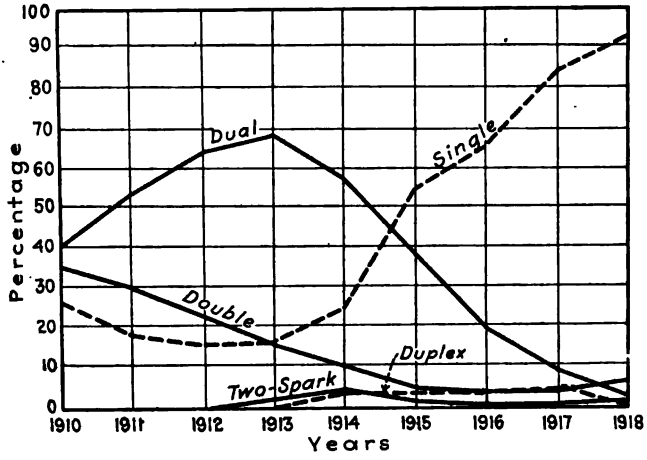

FIG. 10—TRENDS IN THE APPLICATIONS OF IGNITION SYSTEMS

made from forgings, if possible, and if not, from steel castings.

For the vital parts of all trucks and passenger cars forgings will continue to be used. This refers particularly to such parts as axles, steering connections and propeller-shaft parts. This condition has put a check for the time being to the trend toward forgings and stampings, but as soon as the war is over and conditions are relieved it is certain that the forgings and stampings will again come into their own.

A prominent engineer in charge of motor-truck construction in this country states that the following rules guide him in selecting cast-steel, malleable and drop forgings for small brackets, axle parts, etc.:

1 Cast steel is used for the more important structural parts that cannot be forged. It must be made fairly heavy to allow for segregation in the castings and blowholes. It should really be used only for complicated pieces.

2 Drop forgings should be used where very great strength with lightness is required.

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8 Malleable iron should be used for unimportant structural parts, such as spacers, packing pieces and the like; parts where cast iron would probably be strong enough for the actual work to be done, but not strong enough to bear transportation and rough handling.

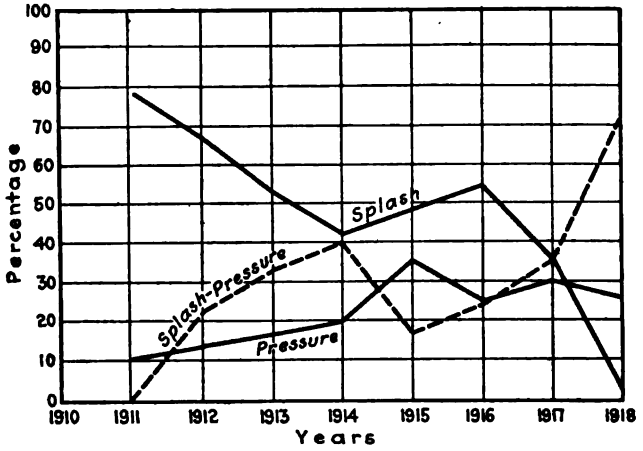

FIG. 11—TRENDS IN AUTOMOBILE LUBRICATION

4 In general, it is a wise rule to use forgings as much as possible, as after the first cost of dies they are usually found to be cheaper and are much more homogeneous.

BODY LINES ALTERED

Bodies generally have been altered only in exterior line. The beveled edge is typical. There has been a slight tendency toward improvement of the front compartment, a thing fought for in some quarters during the past three years. Where there has been a gain of additional space this has usually gone into the front compartment rather than the rear.

The color situation is better now than it was a year ago, although it is not easy to secure the desired pigments even now. A number of brighter colors are now in evidence, showing a tendency toward individuality, particularly among manufacturers who make in the neighborhood of ten to fifteen cars a day.

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One of the tendencies which may be predicted in body design is that the cars of the future, at least those of the touring variety, will be of the two, four or six-passenger type. This is highly desirable from the point of beauty of outline, as it results in a car without the unsightly wheel housings. A two, four or six-passenger body can be swung between the rear wheels, giving a narrow line.

Body Improvement

There is still room for improvement in door fitting, even judging from an inspection of the cars on the floors of the show. This same condition held true last year. Doors will be tight on one side of the body and loose on the other. The shape of the backs of the seats can also be further improved, as the bulge of the back should fit the body. On the closed cars, and particularly the convertible type, a period of uncertainty seems to exist. Manufacturers do not know whether the majority of buyers want a permanent pillar or a removable pillar in the sides of the touring sedan models. This popular car has had a large sale this season, and will continue to have equal success next year, according to present indications. There has been considerable argument in favor of the permanent pillar in that it can be made lighter and at the same time more substantial. This seems to be a matter for education on the part of the purchaser as well as development by the manufacturer.

Manufacturers will do well to scatter wisely throughout the country a few special bodies mounted on their stock chassis. There is always a market for these cars. They serve as very forceful advertising for the company, and can always be sold at an excellent profit. The dealer welcomes a car of this kind for his showroom, and it has been the experience of the companies making these cars that there is always much competition among dealers to secure them.

Author's Summary

Summing up the entire situation, it may be said that we have come to a new era. The improvements in designing which have been traced up to the present time will serve as valuable experience to meet the new period which the war has brought into our national life and which is influencing our demands for transport equipment. The car which will carry its load with the least surplus car

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weight, with good acceleration ability, with no wasted length to add unnecessary weight, with an engine having displacement ample but not too great for the load to be carried, in other words, a vehicle which returns the utmost in performance and comfort for the expenditure, is the one that will most surely satisfy the new demands.

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INDIANAPOLIS SECTION PAPER

SOME FUNDAMENTALS OF ROLLING SUPPORT

By F W GURNEY*

Rollers of one crude form or another have been used from time immemorial to facilitate movement, but it is only within very recent years that the support of revolving parts on rolling bearings has been developed to a state that commands the respect of the engineering world. Rolling support of revolving parts, as distinguished from sliding support, is really a recent innovation in mechanical engineering. As a serious engineering proposition its development is coincident with that of automotive vehicles.

The essential thing accomplished more or less perfectly by ball and roller bearings is the support of moving parts without relative movement between the supporting surfaces in contact. In perfect rolling action there is no sliding contact, only stationary contact. The only movement is where there is no contact. The prime idea of the rolling bearing, stationary contact, is realized through the means of the simplest of mechanical elements, the roller, which gives support to the moving element without the occurrence of any relative movement at the points of support.

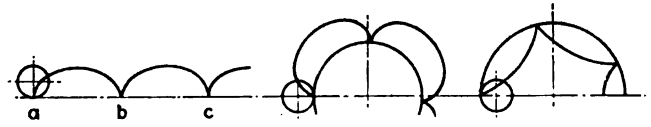

FIG. 1—ILLUSTRATING STATIONARY CONTACT BETWEEN A POINT ON ROLLER SURFACE AND RACEWAY

The fact of stationary contact in rolling support may be graphically shown by tracing the path described by a point in the perimeter of a roller as it rolls over a surface. This is shown in Fig. 1. This point touches the surface on which the roll is rolling at the points *a*, *b*, *c*, coming to a momentary contact and momentary standstill at these points. This imaginary point in a roller describes, when rolling over a flat surface, a cycloidal curve. Such a point on a ball or roller in an ordinary ball or roller bearing describes, with reference to the inner raceway, an epicycloidal curve, and with reference

*Chief Engineer, Gurney Ball Bearing Company

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to the outer raceway a hypocycloidal curve. The action of the elements of a ball or roller bearing is exactly similar to that of a planetary gear.

The bare or basic elements of a rolling bearing are the outer and inner race-rings and the rolling elements, which may be either balls or various types of rollers. This elemental bearing is shown by Fig. 2.

Practical Shortcomings of the Ideal Bearing

Two things are required of a bearing: to carry the most possible load and to carry it with the least possible resistance. The ideal bearing to realize these fundamental requirements would be made up of purely cylindrical elements, an outer raceway of cylindrical form, an inner cylindrical raceway, and cylindrical rolls just fitting the space between these two races. Such elements assembled in absolutely parallel alignment would form a practically perfect bearing. Such a bearing would carry the maximum of load with the extreme minimum of friction.

The ideal bearing suffers certain handicaps under the limitations of actual conditions. The first of these is that the rollers, no matter how nearly perfect they may be, get out of alignment. A roller out of alignment with its race loses its proper bearing or contact with that race. Its contact with the outer race is at its ends, the straight line of the roller being seated upon a concave line of the outer race. Its contact with the inner race is at its middle, the straight line of the roller seating on a convex surface. The roller thus has to endure a transverse load. It becomes a beam. With one well-known roller bearing this difficulty is frankly recognized and met by making the rollers flexible. Second, a roller that is out of alignment with its race, if not forcibly prevented, rolls in a spiral path on that raceway. The lead angle of this spiral path is the same as the angle of disalignment of the roller. The roller must be forcibly prevented from thus rolling, or obviously the bearing comes apart. If we forcibly prevent this worming along of the rollers on their raceways we thereby force them to slide on their seats by the exact amount that they are prevented from worming or spiralling along. This develops considerable friction between the rolls and the controlling means or cages, and very great friction between the rollers and their raceways, for here they are slipping under their full working load, slipping under a biting metal-to-metal contact, where no film of oil can possibly be maintained.

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This difficulty of the roller type of bearing is inherent and fundamental. It is met, with greater or less success, by the use of "cages." The rollers obviously must have a working clearance with the cages, and this clearance inevitably permits a certain amount of misalignment. Furthermore, the sliding friction of the rollers in their

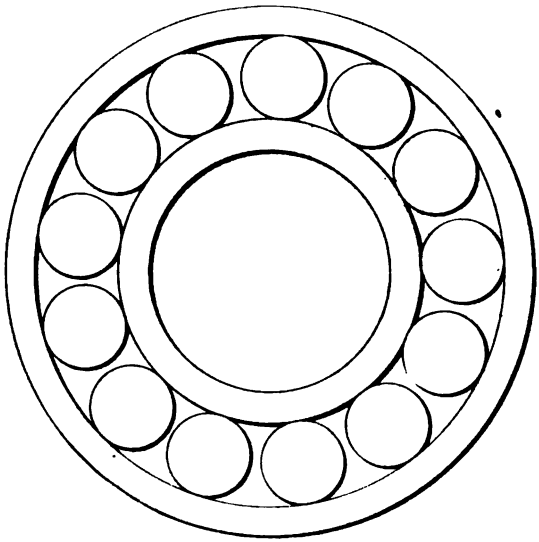

FIG. 2—THE ELEMENTS OF A ROLLING BEARING

cages causes wear, permitting misalignment to increase.

The elemental roller bearing fails to meet common requirements in another particular, in that it has no definite axial position, and does not offer positive resistance to end-thrust. This fault is met by beads and grooves or shoulders or by changing to a tapered construction. The beads, grooves and shoulders add to the frictional handicap and the tendency toward misalignment. Something of the kind, however, is necessary with taper rollers to prevent their backing out from under the load.

BALL BEARING CONSTRUCTION

From these fundamental difficulties, inherent in any kind of bearing using rolls, whether cylindrical or conical, a possibility of escape is indicated by the use of spherical rollers, or balls. By substituting balls for the cylindrical rollers we escape the troubles due to misalignment, but we still have an inoperative bearing, for there is nothing

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to prevent these balls from rolling out at the sides. The obvious and simple remedy for that difficulty is to make grooves for the balls. The great difficulty of assembling confronted the early simple ball bearing. There were three solutions: the cutting of gateways into the raceways from the sides, involving a break in the continuity of the race wall; the method of assembly by eccentric displacement, involving reduction in the number of balls; and the method developed by the author, which I might designate as the sidling-in method, involving neither of these sacrifices.

Probably the first observation made to the disparagement of the ball is that it has less load-supporting capacity, for obviously a cylinder bearing throughout its entire length should support a greater load than a ball with contact only at a point.

This is the great indictment brought against the ball bearing. Advocates of the roller bearing seem to regard it as unanswerable.

In the use of a ball in a bearing, the only stress or load that can be put upon it is a simple load exerted radially, or directly toward its center. It cannot be subjected to tensile, transverse, or shearing stress. The only stress it is ever called upon to resist is pure and simple compression.

The steel that is used in the ball and race rings is a special product developed and used for just this particular purpose. It is an alloy which has been found to possess in the highest degree as yet attainable the properties of hardness and toughness desired to withstand great pressures and resist severe shocks.

The process of ball manufacture has been developed to a point where the day-by-day output of balls runs up into the millions, the balls being true spheres to within a few hundred-thousandths of an inch. The spherical form lends itself to processes, to expedients in production that are not available in the manufacture of any other form. This is due to its absolute symmetry. It presents its surface to the means of fabrication with absolute uniformity in every possible change of position. In this process the sphere is self-developed and self-maintained, and the spherical form is not contingent upon the accuracy of the producing apparatus.

When we couple with this marvelous precision of form a no less remarkable exactness of size whereby a limit of variation not exceeding one ten-thousandth of an inch is realized in the several balls working together in one bearing, some idea is had of the high degree of harmony

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with which the cooperating parts of the ball bearing work together to realize maximum efficiency and capacity.

It is practically impossible to produce rolls of any style with a precision comparable with that of the steel ball such as a number of ball makers are today turning out in great quantities.

Case of Ball Versus Roller

"Line contact versus point contact" by no means epitomizes the case between the ball and roller bearing. Both terms are unfair and misleading, and neither is applicable without qualification to the type which it is intended to characterize. In the ideal roller bearing we have line contact. In the actual roller bearing we merely approximate line contact, more or less closely. This approximation to the ideal alignment and contact moreover is contingent upon the refinement of the crudest part of the bearing, the cage, in which the commercial exigencies forbid any considerable nicety of fitting.

On the other hand, the so-called point contact in the ball bearing is predicated upon assumptions that are unfair and untenable. Point contact assumes that steel is incompressible, and ignores the immensely important matter of race contour. An incompressible sphere resting upon an incompressible plane surface would have contact only at a point. These hypothetical elements referred to are abstractions having no existence in reality. Under the loads of ordinary ball bearing service there is an appreciable area or spot of contact. The extent of this area depends upon three things: the hardness of the materials, the amount of load or pressure, and the character or shape of the surfaces in contact.

The load that a bearing will safely and successfully carry depends obviously upon the character of the steel, the areas of contact, the character of the surfaces and the nature of the contact. The safe working load also is affected by the speed, and these various factors are to a greater or less degree interdependent.

Increasing Area of Contact

There are two ways of increasing the available area of contact between a ball and its raceway: increasing the size of the ball and increasing the curvature of the raceway. Take the case of a disk in contact with a straight line. With the same degree of compression the amount of contact between the circle and the base it rests upon will be in direct proportion to the disk diameter. This

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deals with dimensions in one plane. The sphere deals with dimensions in two planes, and, under the same degree of compression, the contact is in proportion to dimension times dimension, or the square of the diameter.

The areas of contact between two surfaces under the same degree of compression are in inverse proportion to the difference in curvatures of the surfaces. The curvature is the reciprocal of the radius of a curve. When the radius to a curve becomes infinite the curvature becomes zero. When the difference of curvature of two contacting surfaces becomes zero the curves are the same; they fit. It is perfectly logical by changing the raceway to increase the capacity of the ball bearing, by increasing the race curve to approximate more nearly that of the ball. For example: Compare the case of a ball on a plane surface with that of the same ball in a groove with a curvature half or a radius twice that of the ball, Fig. 3. In the second case the difference in curvature is one-half that of the first. By changing from the straight race surface to the race groove of one-half the curvature of the ball the contact area and the capacity of the ball are doubled.

The thing which limits the load supporting capacity of the steel in balls and raceways is the compression of the steel at the places of contact. Failure of the steel simply means that at the place of failure it has been stressed beyond the safe limit. The safe limit of stress is very much less than the elastic limit of the steel.

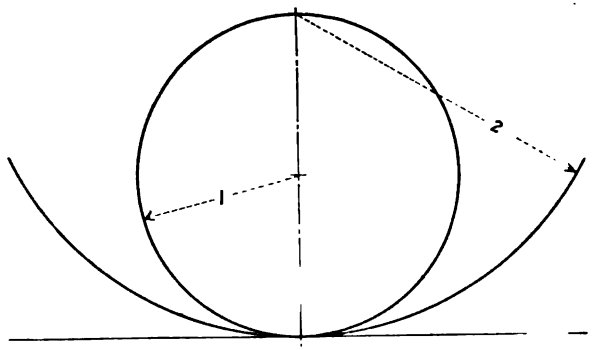

FIG. 3—ILLUSTRATING INCREASE OF CONTACT BY DECREASING RACE RADIUS

There is a limit to the closeness with which the race curvature may approach the curve of the ball. In large bearings for carrying great loads at slow speeds we make the race contour very much closer than in high-speed

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bearings. In one case the chief consideration is load capacity. In the other, the consideration of load is practically negligible, but the reduction of friction is of supreme importance. In ordinary practice the race curvature may be increased to an extent that will multiply the capacity of the ball many times before we incur objectionable friction. Take for instance a ball in a race groove 3 per cent larger than the ball. Under the ordinary working conditions and at the full rated loads the ball rolls along with a contact arc of about 31 deg. This arc is so close to the race curve that it means an exceedingly slight amount of compression, about 0.0003 of the diameter of the ball. Now suppose a shock comes upon the bearing and the load is doubled. We then get an arc of contact of about 37½ deg. and a compression of about 0.00045 the diameter of the ball, still very small. As loads mount up under shock it requires exceedingly slight compression to increase contact areas to the degree necessary to sustain their loads. The sides of the raceways are very close to the balls, to lend their support to the balls when the great shocks come. They are available for these emergencies before the steel has suffered harmful compression. The close race curvature spells safety for the bearing.

Close Race Contour

There is yet another way in which I may make more graphic and impressive the meaning of the close race contour. Let us go back to the flat raceway. To increase our area of contact let us decrease the curvature of the ball. We will convert the spherical roller, for the purposes of this comparison, into a barrel-shaped roller, preserving the diameter of the ball (Fig. 4). Let us give to the curve of this barrel-shaped roller such a radius that the difference in curvature between the roller and the flat raceway will be the same as the difference between the ball and a race curve 4 per cent larger. The meaning of this is that a ball bearing in which the balls roll in raceways 4 per cent larger than the balls, is, as regards load-supporting capacity determined by limits of compression, the equivalent of a roller bearing whose rolls are slightly bulging or crowning in the center, the radius of this swell or bulge being 26 times the radius of the ball. One of the very best roller bearings yet developed has adopted practically this expedient, curving slightly the cross-section of the outer raceway instead of that of the roller.

The ball bearing made with close race contour is in

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effect a roller bearing. In it are realized ample areas of contact which the advocates of the roller bearing have been claiming as their particular possession.

By the application of this practice to ball bearing construction the ball bearing does not lose its unquestioned position as the preeminent anti-friction bearing, but by this means it is securely established as the great capacity bearing, adequate to carry, with the minimum of resistance, practically any loads that may be desired. Ball bearings are being applied with the most gratifying success in positions where the severity of the service makes roller bearings out of the question, and by their use results are realized that had never been attainable with plain bearings. On the other hand, it has frequently been found that under such exceptional loads the friction developed in roller bearings is actually greater than that

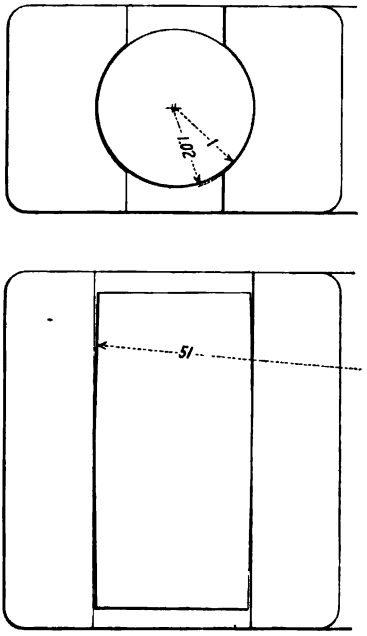

FIG. 4—COMPARISON OF BARREL-SHAPED ROLLER TO BALL IN CLOSE RACE

with plain bearings. This is explained by the fact that with the roller an unpreventable slippage is incurred, which, under the intense pressures involved with metal-to-metal contact and no possible oil film, results in the excessive friction.

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I am a user and an advocate of the roller bearing. There are on the market roller bearings of unquestionable quality and there is a distinctly legitimate field for such bearings. I wish to offer the opinion, however, that in the great majority of cases the requirements may better be taken care of with ball bearings than with roller bearings.

THE DISCUSSION

G. W. SMITH, JR. (M. S. A. E.):—The question of bearing design is usually a compromise and the best bearing may be one or the other, depending upon the inherent capacity. For instance, it is a question whether or not roller bearings are not preferable on account of their smaller outside diameter. In certain cases the rear axle design almost determines the weight of the assembled axle, so the question of limiting conditions often decides which bearing is the better.

MR. GURNEY:—Outside diameter limitations sometimes make it impossible to use ball bearings of adequate capacity. One point little recognized heretofore by most manufacturers is that by the application of close race contour those limitations may usually be made less severe and the desired capacity may be secured without increasing the diameter.

F. E. MOSKOVICS (M. S. A. E.):—I would like to have Mr. Gurney tell what he has found out regarding relative end-thrust and radial loads.

MR. GURNEY:—In the roller bearing, end-thrust is taken care of by taper construction and by shoulders, but not as efficiently as by a ball bearing, with the proper inclination of the axis of rotation of the ball and the contact angle of the ball with its raceway. In this way there is no difficulty in taking care of end-thrust, and ball bearings operate with greater efficiency than roller bearings.

With a full complement of balls in a raceway as deep as we ever use to carry a radial load, the load on all these bearings is practically never great enough to bring contact of the ball up to the edge of the raceway.

C. S. RICKER (M. S. A. E.):—That is about 15 deg. on either side of the plane of the bearings?

MR. GURNEY:—On one side the raceway comes up more than 15 deg. It must be remembered that the arcs of contact I give in my paper are very seldom realized. They are the very extremes. The arc of contact that obtains under ordinary practice is probably not more than 15 or at most 20 deg. The large arcs of contact

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are only instantaneous, taking care of shocks. The ball ordinarily runs in a much narrower path.

C. S. CRAWFORD (M. S. A. E.):—I would like Mr. Gurney to compare ball bearings to taper roller bearings for a hub of the same size; in other words, interchangeable bearings.

MR. GURNEY:—Two prominent firms have conducted some experiments with the Gurney "radio-thrust" bearings and with a popular taper roller bearing. The bearings were of the same diameter, and either bearing would go into the same hub. They equipped a number of cars with roller bearings on one side and ball bearings on the other and ran them on a long endurance test. The chief engineer of one of these firms told me that the average life of the ball bearing in that test was 75 per cent greater than that of the roller bearing.

BEARING LUBRICATION

MR. MOSKOVICS:—I would like to know Mr. Gurney's theory on lubricating bearings.

MR. GURNEY:—If ball or roller bearings could work perfectly there would be no need of lubricants, but there is no such thing as perfection in any of this work. The lubricant is not as important as might be imagined. Ball bearings generally run cooler dry than with a lubricant. We had a case of a ball bearing installation which ran six months without any lubricant in it, and nobody knew the difference. It has been running at 3000 or 4000 revolutions per minute for six months in daily service. The bearing was somewhat worn but not so much as to attract attention. However, lubrication is an exceedingly desirable thing in a ball bearing.

It is necessary to lubricate between the separator and the balls. A film of oil does get between the sliding edges of the balls where the pressure is less intense. Where there is no slippage, there is no need of a lubricant.

LIMITING SPEEDS

D. L. GALLUP (M. S. A. E.):—Is there any condition of speed and load under which ball bearings cannot be used?

MR. GURNEY:—Certainly, but speeds that are impossible with any other bearings can be reached with ball bearings.

MR. GALLUP:—The other day I came across an instance of a manufacturer who changed to plain bearings

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on account of running at high speed. I wondered if it was a mistake on his part.

MR. GURNEY:—The bearings must have been improperly installed. Brown & Sharpe use ball bearings running at 40,000 r.p.m. and doing good work. They found they could not attain this speed on plain bearings.

F. E. MOSKOVICS:—What is a fair speed for the balls?

MR. GURNEY:—The ball speed in the bearings is about 4200 r.p.m. The balls in the bearings are ½ in. diameter. The bearing ran cold.

MR. SMITH:—It has always been my contention that we can make a cheap roller bearing and get away with it fairly well, but that we cannot make a cheap ball bearing and operate.

MR. GURNEY:—Practically Mr. Smith is right. There are some cheap cup and cone bearings which have a real field, but the modern annular ball bearing must be made well, whereas a roller bearing can be made cheaply. In fact, that is one of the principal arguments for the roller bearing.

GRAPHITE LUBRICATION

H. G. MCCOMB (M. S. A. E.):—Manufacturers frown upon graphite grease. I would like to hear an opinion on it.

MR. GURNEY:—We have not experimented a great deal with the use of graphite grease in ball bearings. The Acheson people claim that it is a success. As to the validity of that claim I cannot say. It is important to use a lubricant that is free from any element that will corrode the surface. One point which too few people appreciate is the great importance of keeping bearings clean. There is nothing quite as destructive as dirt, and this does not necessarily come from the outside. There is apt to be considerable abrasion under the bearing, a wearing off of the metal in very minute particles, if there is any vibration of the bearings upon the seat. These particles very soon become oxidized, and oxide of iron in the lubricant is the result. It is a beautiful abrasive, but a very poor lubricant. When the oil becomes red, it should be removed at the first possible moment. Sometimes the wear from the gears produces oxide of iron, which very rapidly destroys the bearings. Bearings are credited with many faults that are not theirs. Too much lubricant should be avoided in high-speed bearings. In the case of very high-speed bearings,

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the lubricant has to be a very light oil, and there should be very little of it.

MR. SMITH:—I would like to get Mr. Gurney's views regarding the commercial efficiency of sub-oiling bearings for live shaft requirements.

MR. GURNEY:—Sub-oiling adds greatly to length of life if it ensures the shaft running on a film of oil with no metal-to-metal contact. It takes slightly more power to run it than with a good ball bearing, but not much more than with an ordinary roller bearing. The great gain in the use of ball bearings in cases of that kind is not so much in saving of power as in saving trouble.

Periodically, without fail, we must go over ordinary boxes to see that they are kept supplied with lubricant. That is not necessary in the case of ball bearings, which can be oiled every six months or once a year or even not that often, and still cause no trouble. A well-designed ball-bearing hanger will contain lubricant enough for over a year, which is a great convenience.

FLEXIBLE FABRIC UNIVERSAL-JOINTS

BY C A SCHELL*

The development of a new product in any industry is generally a gradual evolution from the crude to the highly refined, rather than an abrupt change from one fixed or proved construction to another. This has been so in the refinement of the flexible type of universal-joint and in its introduction to gasoline engine manufacturers, but considering that it has been in practical use in America for only about four years the promulgators of this construction have every reason to feel hopeful. At the present time there are over thirty-five truck and passenger car manufacturers who are using this joint as standard equipment, sixteen of whom use it on the propeller-shaft. All of these installations are giving good service, and their users are enthusiastic about them. Recent converts to this type of construction are the Studebaker Corporation, which uses it on all their models, and the Government, on the Class AA military truck, both placing it between engine and transmission case. The Reo Company, which has used this type of coupling between engine and transmission case successfully

*Chief Engineer, Thermoid Rubber Company

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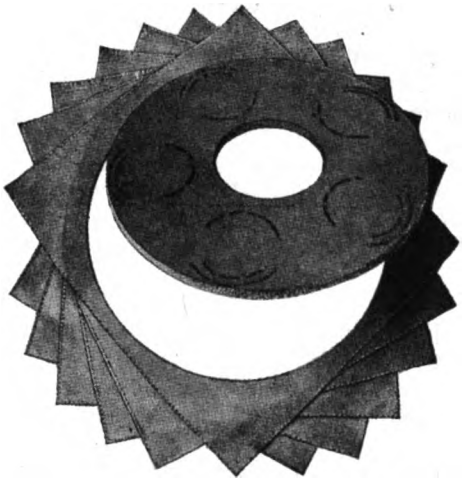
for over four years, has finally adopted it on the axle end of the propeller-shaft. The forward installation was so successful, and was found to have so many advantages over the metal type of joint that it was finally adopted in the rear. The rear propeller-shaft joint generally gives the most trouble, being exposed to much dust and mud, and difficult to lubricate.



SHOWING METHOD OF BUILDING UP A SIX-PLY DISK

Some of the engineers who first experimented obtained discouraging results, in most cases owing to lack of knowledge of the fabrics and friction material required in the construction of the disks, and very often to lack of care and thought in the design of the metal parts. Some of the pioneer manufacturers of the disks are now building them very efficiently, and the design of metal parts is being gradually improved. In some of the first tests conducted it was soon learned that disks made of any ordinary grade of cotton and cheap friction material would not suffice. This type hardened very quickly under severe conditions, and failed just as a tire made very cheaply fails. Very good fabric must be used, and must be placed or woven correctly, the best grade of friction material employed, and the cure must be accurate in order to obtain a strong, elastic, and long-lived disk.

In building our first disks we tried several methods of weaving and of placing the plies of fabric relatively so as to obtain maximum results. We secured very good results from disks woven in six directions or sectors,

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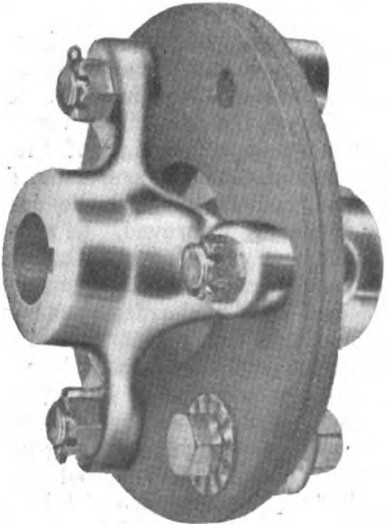
the cotton strands running completely around the disk, and equally good results from disks made up by the Hardy method. The first method proved very expensive, and meant the installation of intricate machinery. We also tried building up the disks by the straight method, or by laying one ply upon the next, with no attention to the direction of the weave, but the results were very discouraging. With this method we got uneven strength and stretching in the different sectors, even though in static tests they were as strong as the Hardy type. When on the road, the sector which happened to be in such relation to a set of bolt holes as to take the greatest part of the load failed very quickly. This is true because cotton is stronger when pulled in certain directions and stretches most when pulled on the bias; naturally with a three-arm type of spider the disk will not offer uniform resistance to the torque and an overload will always be thrown on one sector.

Construction of Hardy Disk

We finally adopted the Hardy disk, which is made by twisting the plies in consecutive fashion so that the disk is uniform in strength and stretch from any one portion to another, and when made of a good fabric has a tensile strength of 3000 lb. per sq. in. The Hardy disk had been used in England for several years prior



FLEXIBLE FABRIC UNIVERSAL-JOINT WITH TWO DISKS

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to our experience with it, and is being used on several of the chassis over there at the present time. Of course, disks of other constructions can and are being used, but in starting out we aimed to get the greatest strength and longevity that could be obtained in a given space.

We also aimed to get considerable elasticity in order to benefit all the chassis parts as much as possible, as the cushioning effect of elastic disks adds greatly to the life of the shafting, bearings, gears, transmission, clutch and even the engine.

After experimenting considerably with disks we went thoroughly into the metal parts of the joint and are conducting interesting tests at the present time. In testing some of the first assemblies that had given any trouble, we found that it was often due to the lack of care in designing the metal parts, and we have seen excellent disks fail owing solely to the use of poor friction washers. Most of the first washers were made too thin and allowed the fabric to pull out under the surface, throwing the load on the bolt holes until failure took place. The washers should be made thick enough to resist cupping and should be fluted or knurled like a clutch-pedal in order to grip the fabric snugly and drive by friction. It is also good practice to groove or knurl the driving side of the spider bosses.

It must be remembered that the disks are comparatively soft, and that the volume or section held tightly under the washer face is depended upon to do the work; a disk made of the most expensive materials will fail unless efficient washers are used.

Life of Flexible Fabric Joints

We have installations on the road which have run over 60,000 miles over a period of four years, and look good for some time to come. These consist of well-made disks of good fabric and well-designed metal parts, and we are satisfied that a well-balanced assembly will last as long as most present-day chassis. Even should it become necessary to install a new set of disks once during the life of the vehicle, the benefit derived by the rest of the chassis in the meantime is worth many times this expenditure. I believe the saving created by the use of a flexible coupling in the transmission line is not fully appreciated.

At the present time we are engaged in some laboratory work which will be of interest to the engineer. We have two well-known chassis, and with the use of special ap-

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paratus will attempt to show the failure points of certain parts under severe shock with and without the installation of flexible disks in the transmission line. This is a difficult task and will require time to be put in conclusive form, but from present observation we are sure the results will be almost astounding. The greatest benefit is obtained on trucks with solid tires, when chains are employed. It is not uncommon to see a truck with two or three pieces of chain wrapped around the rear wheel started with an unmerciful jerk by a rough driver, and in such cases a small shock absorber somewhere in the drive is an advantage. In a case like this a good disk-joint will allow about 5 deg. wind, which means a considerable time element; the cushioning effect of the Hotchkiss drive is small in comparison, especially on trucks with huge rear springs.

Use with Hotchkiss Drive

One of the strongest arguments for the use of the Hotchkiss drive is the flexibility of transmission; therefore, with the installation of the flexible coupling in addition a still greater cushioning effect is obtained; of course, this cushioning can be obtained even on fixed axles with the coupling, and, as stated above, to a much greater extent than with the Hotchkiss drive itself.

After experimenting and going thoroughly into the proposition, both in the laboratory and on the road, the Studebaker Corporation has adopted the flexible disk as a clutch-to-transmission coupling on all of its new chassis. The engine in the large Studebaker chassis develops a torque of about 2400 lb.-in., and in this case a single disk 7½ by $^{13}/_{32}$ in., consisting of 10 plies of fabric, is used. By actual test this disk will fail at a torque between 12,000 and 13,000 lb.-in., pulling through 1⅜-in. washers. This complete assembly, compared with a metal type for the same place, is lighter, requires no attention, eliminates backlash and noise and adds greatly to the life and safety factor of all metal parts. In the Studebaker disks the bolt holes are on a 5⅝-in. radius with 1⅜-in. washers. With the use of a four-arm spider, this gives an effective driving section of 2¼ sq. in. of fabric, having a tensile strength of 3000 lb. per sq. in. Therefore the disk will carry a 6750-lb. load at the bolt-circle radius. The torque at this distance from the center equals 850 lb., consequently this assembly has a very high factor of safety.

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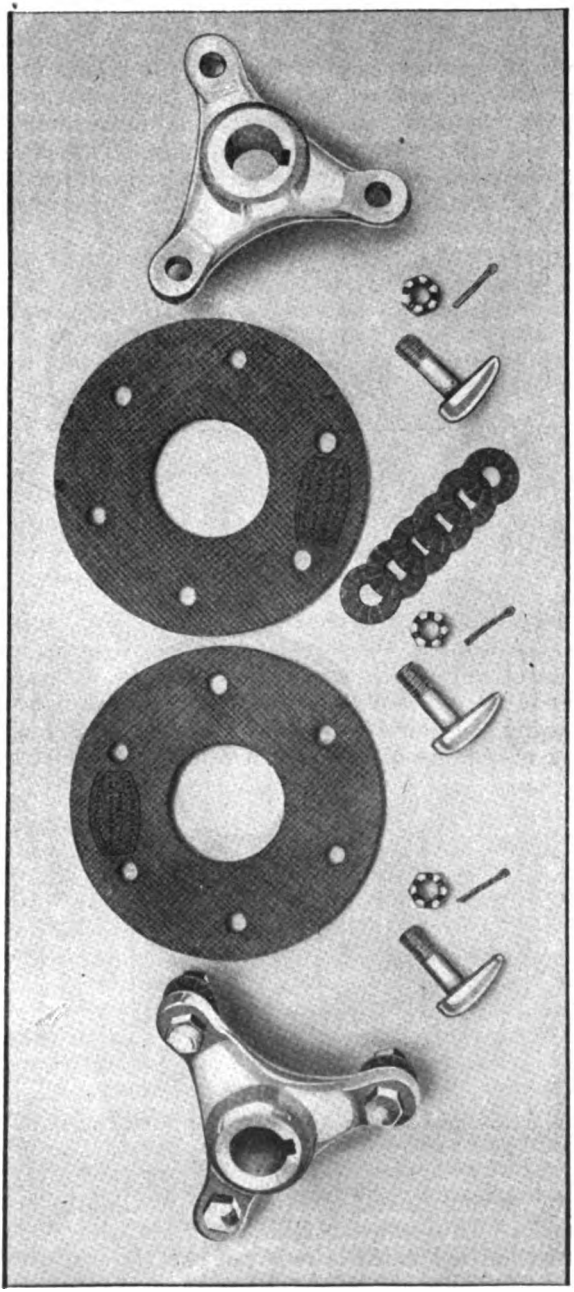

COMPONENT PARTS OF A FLEXIBLE FABRIC UNIVERSAL-JOINT

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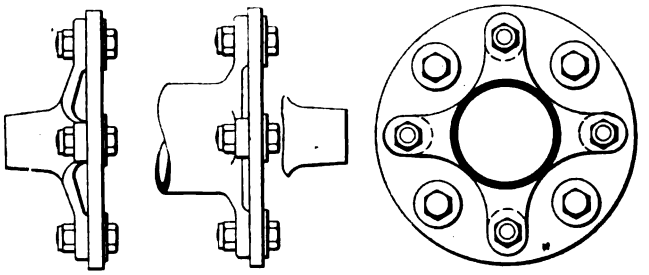
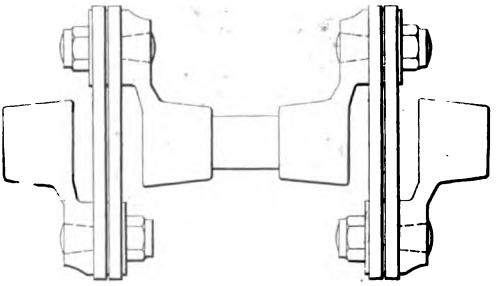
Class AA Military Trucks

Two of the engineers on the committee designing the Class AA military truck have had previous experience with flexible joints, and, seeing them successfully used, have laid out a well-balanced assembly. Two disks per joint are used, of 6½ by 5/16-in. material with a 1¾-in.



FOUR-ARM TYPE OF COUPLING FOR USE BETWEEN ENGINE AND TRANSMISSION

fluted washer, with three-arm spiders. The engine develops 1800 lb.-in. torque; hence this will prove a very successful unit, with an ample factor of safety. For engine-to-transmission-case couplings, many designers are now using the four-arm spider similar to the Studebaker, taking advantage of the fourth arm for extra strength,



TYPE OF FLEXIBLE FABRIC UNIVERSAL-JOINT USED ON CLASS AA MILITARY TRUCKS

thereby saving fabric and making a slightly cheaper assembly. This construction decreases the flexibility slightly, but not much is required as only chassis variation and weaving are to be taken care of at this point.

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Use Between Engine and Transmission

For installation between engine and transmission the flexible coupling eliminates all bearings and splines, thereby avoiding the possibility of noise and doing away with the inconvenience of greasing. In addition it reduces severe shocks, makes easier the shifting of gears and owing to the above qualities adds greatly to the life and safety factor of all metal parts.

Propeller-Shaft Joints

In laying out propeller-shaft joints many factors besides torsional load are to be considered, the angle and amount of lateral movement being the most important. On account of these conditions the three-arm type is the most practical and is the only one now used. The diameter and thickness of the disks are partly dependent on the lateral movement and angularity, and should have capacity enough to take care of the increased torque while in low gear. The average light six now in vogue in this country requires a three-disk coupling from 7 to 8 in. diameter, either ¼ or 5/16 in. thick.

An installation of this size takes care of the average angularity and requires no spline for lateral movement. As the trend of design is toward flat springs and straight propeller-shafts, it will be possible to equip more cars each year, as this element cuts down the work of the disks to a minimum, making conditions ideal for this type.

Contrary to general opinion, the thrust required to stretch the disks to take care of lateral axle movement is very slight. The variation is taken up in two sets of disks, and if a total of ¾ in. is required, then each coupling must move only 3/16 in. either way; this requires less than 40 lb. total pressure on the average passenger car. The load imposed on adjacent bearings due to tipping is also very slight but is dependent on the angle. We have several rear assemblies now running over 40,000 miles which have an average shaft angle of 6 deg., with a maximum of 10 deg., but in going higher than this the outside diameter should be very closely watched.

Many of the first engineers to try out propeller-shaft couplings had considerable trouble with vibration. This was very often due to poor machine work or a bad combination of disks. Several light cars, both fours and sixes, with fairly high-speed engines, are now using the rear installations, and when the assembly is properly ma-

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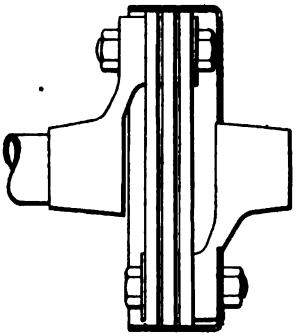
chined experience no trouble of this kind. The vibration led to the installation of centering devices, and some of these were not very successful. When extremely high speeds along with large angles are encountered, a good centering device will help to cut down the small amount of vibration which may develop; however, on the average light six of the present period we have not encountered any serious trouble of this sort, and one very popular light four equipped with a high-speed engine shows none whatsoever at 50 m.p.h. One of the large truck makers using this construction puts a pressed-steel disk over the fixed spider fitting closely around the disk thereby causing the disks or floating spiders to rotate truly within the shell. The centering device, we believe, can be used to advantage under extreme conditions, but on the average car is unnecessary. Of course, when a large angle and extremely high speed are combined the shaft will rotate



STEEL DRUM USED AS A CENTERING DEVICE

elliptically—to a greater extent than when conditions are more moderate. Today several light sixes are being equipped with three-disk joints which we consider average as regards speed and angularity of drive. An average shaft angle is 5 deg. with 4½: 1 axle reduction and a total lateral axle movement of ¾ in. None of these installations have a centering device or a spline, but they are giving good results. For slower speeds larger angles can be used. If the hole in the center of the disk is not too large the disk will be very strong in a direction outwardly from the center, and still be flexible laterally. Naturally the floating spiders will be kept running in a very small ellipse, thereby keeping the assembly from whipping. We do not wish to convey the idea that the flexible joint can be mounted on any

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and every chassis, as such is not the case. We have them on the final drives of four-ton worm-driven trucks, and on fairly high-speed light touring cars. When the conditions are right, the angle not too great, or the speed not too high combined with a large angle, the results are excellent, saving the designer much worry about noise and lubrication troubles.

We have found it unnecessary to pay particular attention to the position of pinion-shaft in relation to transmission main-shaft. Due to the stretching of the fabric, practically no variation in the speed of the pinion-shaft occurs on the average installation.

We have a Hotchkiss-drive chassis at our factory which contains a very good low-speed engine, and has a propeller-shaft mounted at about 8 deg. with the car light. With the metal joint installed, the pinion and transmission shafts must be lined up very carefully in order to get smooth low-speed performance and eliminate a slight jerking of the car. With the use of the flexible joints, we have placed these shafts in every conceivable relation to each other and still get smooth action.

A complete shaft assembly with well-designed metal parts weighs about 80 per cent as much as a metal joint of similar capacity. It requires no attention, cuts out backlash, adds to the safety factor of the other chassis parts and acts as a sound insulator between axle and frame. We feel that our confidence in this type of construction has been justified by the success its adherents are having with it. We predict it will be almost universally used between engine and transmission case within the next few years, and that for propeller-shaft installation it will give the metal type a very stiff argument for predominance.

THE DISCUSSION

H. G. McCOMB (M. S. A. E.):—About how long ago were these first used? Were they brought out first in England and who used them first?

MR. SCHELL:—I believe they were first used on the Isotta-Fraschini in 1911 and were used in England soon after.

MR. McCOMB:—Does the Fifth Avenue Coach Company use them?

MR. SCHELL:—No, it does not.

MAX H. THOMS (M. S. A. E.):—When several disks make up a joint, is a fluted washer used between adjoining disks?

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MR. SCHELL:—A fluted or knurled washer is used between each disk.

Effect of Oil and Heat

MR. THOMS:—What effect do oil and heat have on these joints?

MR. SCHELL:—An excess of oil will eventually destroy cotton or rubber but on the average chassis the disks do not come in contact with enough to harm them. Under very extreme heat they will become brittle and eventually crack.

LIEUTENANT MEYERS:—In the trucks used for road work by the Barrett Company, the universal is so situated that it is exposed to the tar, which is, of course, a much more solvent agent than oil. The universals have been tested out carefully—I think forty-eight were tested in a bath of tar around 400 deg.—and there was no effect, practically, in disintegration of the fabric; but one of those universals was mounted over the muffler, and that, in the course of two or three months, showed some disintegration, owing to the heat. That was the only trouble experienced.

Material Used

MR. THOMS:—Is the material used like brake lining—that is, interwoven with fine-meshed brass wire?

MR. SCHELL:—The material used in the construction of the disks is made of selected cotton duck and high-grade rubber. The duck is made of three grades of cotton, Sea Island, Egyptian, and the best grade of American, called combed peeler. Of the three the Sea Island is superior.

MR. MCCOMB:—Has that movement of about 5 deg. really an important effect on the operation of the car? Some years ago there was brought out a flexible shaft composed of a number of bars that were actually rotationally flexible. It seems to me that if there had been much in that principle, it would have been developed. There is only about 1/72 of a turn, so far as rotation is concerned, due to 5 deg. action of this joint.

MR. SCHELL:—The variation in speed depends upon the angle of the propeller-shaft. On a chassis at our factory which has a shaft mounted at 8 deg. we get a noticeable variation at very low speeds when using metal joints which are improperly mounted. Because of the stretching of the fabric in the disks, this does not occur when flexible joints are installed.

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MR. MCCOMB:—Is the jerking effect noticed only at low speeds?

MR. SCHELL:—The variation in propeller-shaft speed is naturally there at all car speeds but is noticeable only when running very slowly. Most of the metal joint manufacturers recommend the mounting of their joints in certain relations to one another in order to overcome this speed variation.

Maximum Life

E. B. REESER (A. S. A. E.):—Have these joints been in use long enough to determine the maximum life, in mileage?

MR. SCHELL:—The Reo Company has been using a forward installation for about 3½ years and I know of some of these cars that have run over 60,000 miles in this period. Several motor trucks have used the rear installation for 3 years over a distance of about 40,000 miles.

C. S. MOTT (A. S. A. E.):—Can the weight of the driving shaft be decreased by the use of joints?

MR. SCHELL:—A well-designed flexible joint will weigh considerably less than a metal joint of similar capacity.

MR. MOTT:—We had considerable trouble because both joints were operative at all times. There was excessive spring action and I believe the slippage on the spline was from ⅜ to ⅞ inch. Has anyone had experience running with a gear ratio of say 5 to 1 at fairly high speeds?

MR. SCHELL:—We have installations on light sixes that have gear reductions of 4¾ to 1, the engines turning over at 2500 r.p.m., and they are very successful.

MR. MOTT:—It has not been tried out on anything much heavier, then, at high speed; for instance, on an eight-cylinder, with fairly good torque, from 1½ up to 60 miles, with drive shaft 56 in. long?

MR. SCHELL:—One western company is using the installation on a twelve-cylinder model that weighs about 3500 lb. The engine is of fairly high speed and develops about 60 hp. The rear axle ratio is close to 5 to 1. The drive-shaft is about 59 in. over-all.

MR. MOTT:—On this particular job, the springs have considerable bow. Is a spline slip advocated with a Hotchkiss drive?

MR. SCHELL:—If the lateral movement is not over ⅞ in., I would not use a spline. The diameter of the disks will be in proportion to the amount of power transmitted. What was the torque capacity of the engine?

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MR. MOTT:—I do not know at exactly what it was rated, although it gave 56 hp. at 2600 r.p.m.

C. S. CRAWFORD (M. S. A. E.):—That would be about 150 lb.

MR. SCHELL:—A 7½-in. disk would be appropriate for this chassis.

Proper Number of Bolts

MR. THOMS:—If there is any play to be taken over by the universal-joint, would it be preferable for the joint to have four driving bolts, or should it have six? With so much play the material is stretched far beyond the safe load or safe strain.

MR. SCHELL:—All the propeller-shaft installations today incorporate a three-armed spider which is flexible enough to take the lateral movement and gives three driving sectors. If the two-armed spider is used, naturally only two-thirds of the effective material is utilized and the disk must be larger in proportion to transmit a given load. Where the lateral movement is very large it would be advisable to use a spline in connection with a three-armed spider rather than adopt two-armed construction.

J. E. PADGETT:—Is there any difference due to continuity of impulses in the life of those used with four-cylinder engines, and those with a greater number of cylinders?

MR. SCHELL:—Naturally with a four-cylinder engine the torque is not as even and this is generally allowed for in laying out an installation. We have noticed no great difference in the life of an assembly in connection with different numbers of cylinders.

MR. MCCOMB:—For the same load, how much larger in diameter are the fabric joints than steel joints of standard construction?

MR. SCHELL:—The flexible coupling generally requires from one to two inches more space than a metal joint of the same capacity.

Lubrication of Metal Universal-Joints

MR. CRAWFORD:—I think the tendency is always to use the smallest thing on the data sheet. Conditions are constantly changing and when figuring these things out, it is best to have the engineer on the ground to work out the problem with one. We have one job that has been run a little over 10,000 miles, with 6½-in. drive. A larger drive was recommended to us, but we wanted to see the smaller

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one in operation first. It has two disks and a three-arm spider. The lateral movement on our job, however, from the first rebound to the bumpers against the frame, was only ⅜ in. The springs are practically flat. The maximum angle is about 7 deg. That car ran on the Speedway for most of the 10,000 miles at an average engine speed of 3200 r.p.m. Our job is not altogether immune from oil, and I know the disks are soaked with it, but we have never noticed any disintegration due to the oil at high engine speed. We can run up to 3500 or 3600 r.p.m. in second-gear, at which speed there is a bad whip in the drive. However, I have never seen a metal type universal-joint that would stand a half day's running on the track and hold the lubricant. The joint becomes so hot at that speed that it burns out all the lubricant we have ever tried. The lubricant may stay in the housing, but the joint becomes so hot it melts the lubricant around the housing and the pins there then start to gall, unless the pins are fitted very loosely, and then there is much noise. A pilot might be used for the central part of the drive to support the shaft. I think that could be worked out well, without getting any wear and without any lubricant, especially where there is very slight slippage. I think the lubricating problem is the one thing against the metal joint. We have had more trouble with that than with anything else.

MR. THOMS:—Nevertheless, they run racing cars with those joints.

MR. CRAWFORD:—I know they do, but I do not think they run them at these engine speeds, and in racing cars the drive is as straight as possible. Racing cars are lashed down and do not bounce around. I think conditions differ with passenger vehicles. A racing car has not the load that a touring car has, nor the top and windshield.

MR. SCHELL:—In regard to lubrication, we carried on a very unique experiment in New York City about four weeks ago. Several of us were of different opinions as to the service the metal universal-joint generally gets on the average car. I made a wager that eight of every ten owners who drove their own cars did not know when their joints had last been greased or inspected. We left the club and approached the first ten owners of cars whom we happened to meet, picking out men who did their own driving. Instead of eight not knowing anything about their joints we found that the entire ten did not know. We asked several why these had not been taken care of and some said they did not know that joints had to be

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greased. I asked one man if he ever read his instruction book; he very intelligently informed me that people who write instruction books generally know nothing about automobiles. One informed me that he did not know he had an instruction book. Most metal universal joints, if designed with ample bearing surfaces and given proper attention, are successful; but this little experiment showed us that the average car is not given a square deal.

MR. THOMS:—What is the comparative power loss in driving through an angle with this type of coupling and a metal coupling?

MR. SCHELL:—We are carrying on experiments to determine that at the present time and shall know exactly from actual tests in the very near future.

MR. MCCOMB:—Why have not more high-grade cars adopted this type of fabric joint?

MR. SCHELL:—New and advantageous developments are not always adopted by builders of high-priced cars, although several high-priced trucks and passenger cars have been using them for some time.

MR. MCCOMB:—Are they not being used between the engine and transmission only?

MR. SCHELL:—A number of cars are using them on the propeller-shaft.

Comparative Cost

MR. MCCOMB:—They have been used so much in motorboat service, that it seems odd more automobile manufacturers have not come to use them. Has the price operated against it?

MR. SCHELL:—The price at the present time is about that of a good metal universal-joint installation. This is kept high due to the abnormal cost of cotton used in the manufacturing the disks.

MR. CRAWFORD:—If the car manufacturer would make the chassis changes to take care of the conditions under which this joint should be used, more of them than of the other type would probably be used, and if he would do the same thing for the metal joint, he would no doubt get much better results. I do not believe the application of universal-joints to cars has had the serious consideration that it should have.

H. A. VOST:—I notice that in several joints spoken of three 5/16-in. disks were used. Can two 3/8-in. disks be substituted?

MR. SCHELL:—Two could be used but in using the three

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two additional frictional driving faces are obtained. Naturally, with a lower given torque less surface friction is required for driving.

MR. VOST:—For the man who has a light car, and two 5/16-in. disks, the use of three 1/4-in. disks would be recommended?

MR. SCHELL:—In such a case I think the three 1/4-in. disks would give better results.

MR. VOST:—I would like to know the tolerances allowed in fitting the disks on the studs?

Bolt Holes

MR. SCHELL:—The bolt holes in the disks are made the same size as the bolts, which must be forced slightly when assembling.

MR. THOMS:—Are the holes punched midway between the outer and inner edges of the disks?

MR. SCHELL:—We do not put the bolt holes in the center of the fabric. They are slightly out from this as the factor of flexibility must be taken into consideration on this point.

MR. THOMS:—Have not some elliptical washers been used?

MR. SCHELL:—We have used elliptical washers and still use them where not much angularity or lateral movement is needed but, on the rear installations, are getting best results with the round fluted washer.

MR. SMITH:—Is it not a fact that on high-speed engines the disks loosen up, so that the two ends of the rear shaft rebound from time to time, causing considerable vibration?

MR. SCHELL:—With large angles and extremely high speeds we have had some vibrational trouble. This can be overcome by the use of a centering device but, as stated in my paper, on the average chassis we find this unnecessary.

Disk Diameters

MR. SMITH:—My own experience, on a small car, is that the disk should be greater in diameter in proportion to the horsepower used. Is that not true? For instance, a 5½ or 6-in. disk is used on a 30-hp. engine and possibly a 7 to 7½-in. disk on a 50-hp. engine. That may show that the relatively larger diameter gives better results on the smaller car.

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MR. SCHELL:—The diameter of the disk is generally in direct proportion to the power transmitted and we have been getting just as good results with 7½-in. or 8-in. disks as we have with the smaller ones. Of course the lateral slip and angle must not be neglected in designing a joint. Many of those who experience this trouble do not pay very strict attention to machining the parts accurately. This is as essential as in machining a metal joint. Some of the first assemblies were equipped with disks which were too flexible and were built with too large a center hole; this made the disk too weak to overcome the centrifugal action due to working on an angle.

TRACTOR FRICTION TRANSMISSIONS

BY CHARLES A. TRASK*

The type of friction drive with which this paper chiefly deals consists of two wheels—one fiber and the other metal—operated by having their rim surfaces pressed together along a line of contact and transmitting a pull due to the resistance of the driven member to sliding or slipping on the driver. If we know the pressure and the coefficient of friction in such a drive, then the product of the two will give the tangential pull. With any given drive the pull can be resolved into foot-pounds per minute, and dividing by 33,000 the horsepower transmitted can be determined. Thus:

$$Hp. = \frac{\text{(tangential pull, lb.)} \times \text{(tang. speed, ft. per min.)}}{33{,}000}$$

For a wheel D = Diameter in inches.

N = Revolutions per minute.

$$Hp. = \frac{\text{(tang. pull, lb.)} \times \pi\, D\, N}{33{,}000 \times 12}$$

$$= \text{(tang. pull, lb.)} \times 0.000{,}007{,}938\, D\, N$$

But tangential pull $= f \times P$

In which f = Coefficient of friction.

P = Pressure, lb.

In the most popular combination of friction materials (one wheel of cast iron and the other of tarred fiber) the following recommendations are made:

For spur, bevel, and miter drives (where faces have motion of true rolling contact) $f = 0.277$.

*Experimental Engineer, Rockwood Manufacturing Company

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For variable-speed disk friction drives (where faces do not follow true rolling contact) $f = 0.324$.

Also it is recommended that a pressure not to exceed 250 lb. be used for each 1 in. width of face of fiber.

Then P (total pressure) $= 250\ w$, when w = width of face in inches.

Hence for spur, bevel, and miter drives:

$$Hp. = (0.277 \times 250\ w) \times (0.000,007,938\ D\ N) = 0.000,55\ D\ w\ N \qquad (1)$$

For disk drives:

$$Hp. = (0.323 \times 250\ w) \times (0.000,007,938\ D\ N) = 0.000,64\ D\ w\ N \qquad (2)$$

As the power capable of being transmitted in friction drives varies directly with the pressure, it would be ideal to use friction materials with high compression strength. But it has been found that as a fiber material is increased in density to carry greater pressure, the coefficient of friction decreases. Therefore, as pressure multiplied by coefficient represents pull and as it is impossible to increase the value of one except at the loss of the other, no final gain results. *Pressure-withstanding quality of the fiber friction material is therefore the limiting factor in the friction drive.* If pressure is too high the structure of the fiber gradually yields and breaks down. This breaking down or disintegration at high speeds results in a charring or heating of the fiber; it becomes "brashy" and goes to pieces. Everyone knows the result of running on an under-inflated or over-loaded automobile tire; the wall fabric becomes sheared from the rubber, heat is generated and soon a new tire is needed. The effect of excessive pressure on a fiber friction wheel is very similar. It is absolutely necessary that pressures be kept within reasonable limits. The pressure of 250 lb. per in. of face width of fiber, given above, represents only about one-sixth to one-tenth the pressure that would break down the tarred fiber wheel for a short period of running, but for average long working periods 250 lb. is all that is safe, and even that may be high for rim speeds of from 6000 to 10,000 ft. per minute.

Application of Pressure

The simplest and best method of applying the pressure between the friction surfaces would be through a foot pedal, or a lever that could be drawn up until the pressure is just sufficient to handle the load and then retained by suitable ratchet or pawl at any desired posi-

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tion. In every friction drive a greater pressure is required to start a load, owing to force required for acceleration and to the fact that the coefficient is less (100 per cent slip) than under normal running conditions. Also there will be times when overloads must be carried for a short time, as on short heavy grades, in sand, or when plows are dropped. The single pedal or lever permits the operator to readily meet all these conditions—just as a hand-controlled belt-tightener might be desirable to meet short overloads, so that the belt is not always operated under highly destructive tension.

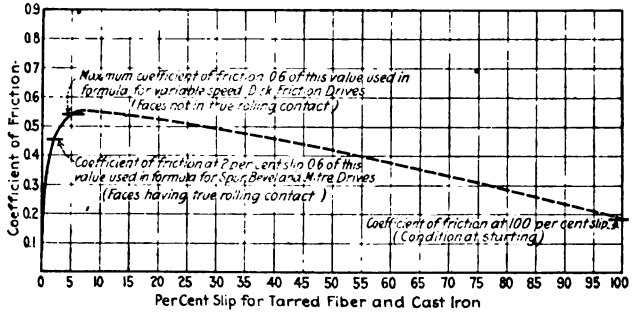
Coefficient of Friction
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
Maximum coefficient of friction 0.6 of this value used in formula for variable speed Disk Friction Drives (Faces not in true rolling contact)
Coefficient of friction at 2 per cent slip 0.6 of this value used in formula for Spur, Bevel and Mitre Drives (Faces having true rolling contact)
Coefficient of friction at 100 per cent slip (Condition at starting)
0 5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100
Per Cent Slip for Tarred Fiber and Cast Iron

FIG. 1—RELATION BETWEEN COEFFICIENT OF FRICTION AND SLIP IN A FRICTION TRANSMISSION

Industrial and farm vehicles are frequently placed in the hands of those whose muscular development exceeds their mental development and it is necessary to either train the driver in order to obtain approximately the correct pressure or make the drive foolproof to guard against his lack of mechanical sense. Hence for tractor service it seems advisable to consider the independent application of a fair operating pressure under normal load conditions and then provide auxiliary means to obtain greater pressure for short periods, as the operator may find that the conditions warrant. This arrangement can be carried out through a lever or arm controlled by a spring, which will always operate the friction members under a fair average pressure when engaged (like the ordinary automobile clutch), and an auxiliary lever or pedal provided without ratchet or stop, to be effective only as long as the operator exerts pressure through them. In any event, while sufficient pressure is necessary to operate the friction drive without undue slip, practically all the trouble encountered is caused by excessive pressure, and some guard must be provided against its

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continual use. Like the belt drive, the ordinary friction drive operates with only a small percentage of slip, and any operator soon becomes able to tell if this becomes excessive.

Slippage

Fig. 1 shows a curve representing the typical relation existing between coefficient of friction and the slip between the driving and the driven wheels. This relation is quite similar to that holding true for the ordinary pulley and belt drive. Starting at zero per cent slip the coefficient of friction (or the resistance to slipping) increases and reaches a maximum at about 6 per cent slip.

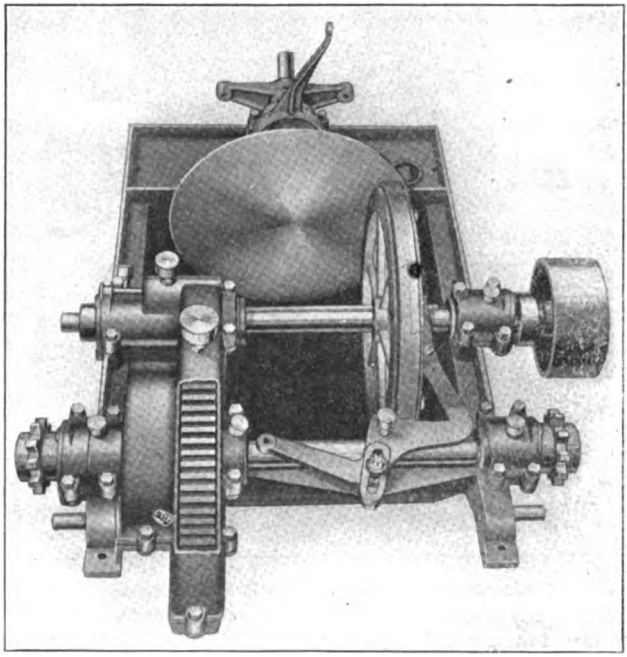

Fig. 2—Disk Friction Drive of the Elgin Tractor

This is the point of maximum pull; from then on the coefficient decreases again until at 100 per cent slip it is only about one-third that at 6 per cent. The common practice is to design a belt drive for a value of the coefficient corresponding to 2 per cent slip. Likewise 2 per cent slip is used in designing friction drives having spur, bevel, or miter shapes in which the faces follow mo-

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tions of true rolling contact. At 2 per cent slip on the curve the value of the coefficient of friction as shown is about 0.462. However, the curve represents values under ideal conditions as obtained in laboratory test. To provide a fair factor of safety for average working conditions the actual value is reduced 40 per cent, giving as a conservative working value of the coefficient of friction, 0.277, as used in the calculations for formula (1). In

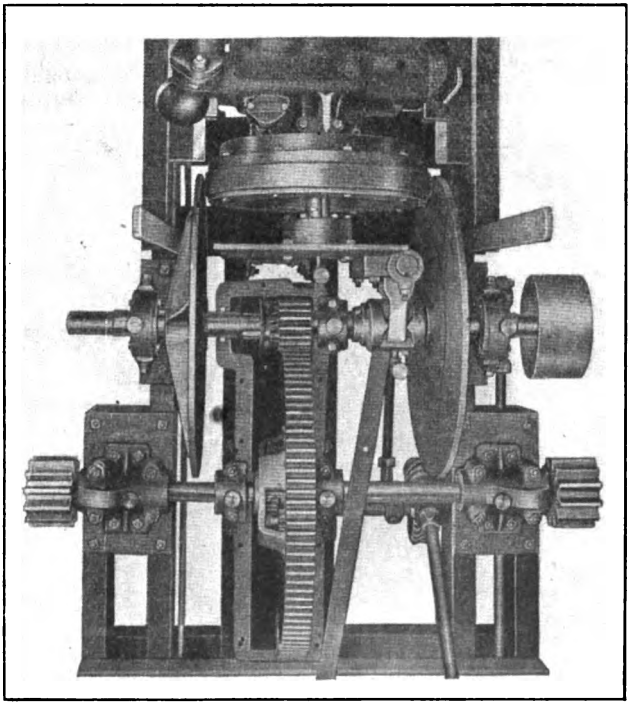

FIG. 3—DOUBLE-DISK TRANSMISSION OF THE HEIDER TRACTOR

disk friction variable speed drives, it is impossible to design for a fixed percentage of slip between the driving and the driven members, as a slight twisting action always occurs on the wheel face, owing to the lack of true rolling contact, and the actual slip varies with the different positions of the wheel on the disk. The tendency in this arrangement of drive is for the slip to increase until, for the different points on the face, the total average value corresponds to the value of the coefficient just sufficient to carry the load under the pressure being used. Natu-

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rally for greatest transmitting capacity this is represented by the values at the peak of the curve. Therefore, it is customary to design disk drives on the basis of average maximum values. In the case of the curves shown in Fig. 1 a fair average maximum value of the coefficient is 0.50 and 60 per cent of this, taken as a fair working value, is 0.324, as used in formula (2).

Between values of 6 and 100 per cent slip (shown by the broken line) the relations on the curve are assumed. These no doubt are approximately correct, and in any event the values within these points do not materially affect practical operating conditions. The reduced value of the coefficient at 100 per cent slip explains why for a given fixed load a proportionally higher pressure is required at starting, and also why this condition is independent of the pull needed to accelerate the load from rest.

Operating Speeds

Since power is merely the rate of doing work, and any given friction has a fixed maximum tangential pull corresponding to the pressure of contact and the coefficient of friction, the power-transmitting capacity of friction drives varies directly with the speed of operation. There appears to be no particular limit to the safe operating speeds of friction wheels, such drives having been operated at rim speeds up to 13,000 ft. per minute, but for average conditions the surface speed may be anything up to 6000 or 8000 ft. per minute.

Design and Installation

In the case of variable speed disk drives, these surface speeds should be obtained by the use of disks of large diameter rather than by high rotating speeds, so that in low-gear position the friction wheel will reach such a point on the disk that the surface speed will be moderate. For this reason, the operator should slow down his engine when starting the load so that when the surfaces are engaged there will be less slip to overcome and less wear of the friction surfaces as a result.

In mounting friction transmissions, the speeds usually being high, it is well to see that all parts are properly balanced. Ball or roller bearings, both radial and thrust, of the self-aligning type, or bearings mounted in self-aligning housings, should be used to compensate for possible springing or deflections of the shafts, and for distortions owing to road inequalities. This point is important as otherwise bearing trouble may develop.

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The average designer appears to consider that his shafts are called upon to transmit only a given torque. In friction drives they must also carry the pressure and maintain the correct alignment of the faces of the friction members. Actual operating conditions are not those of

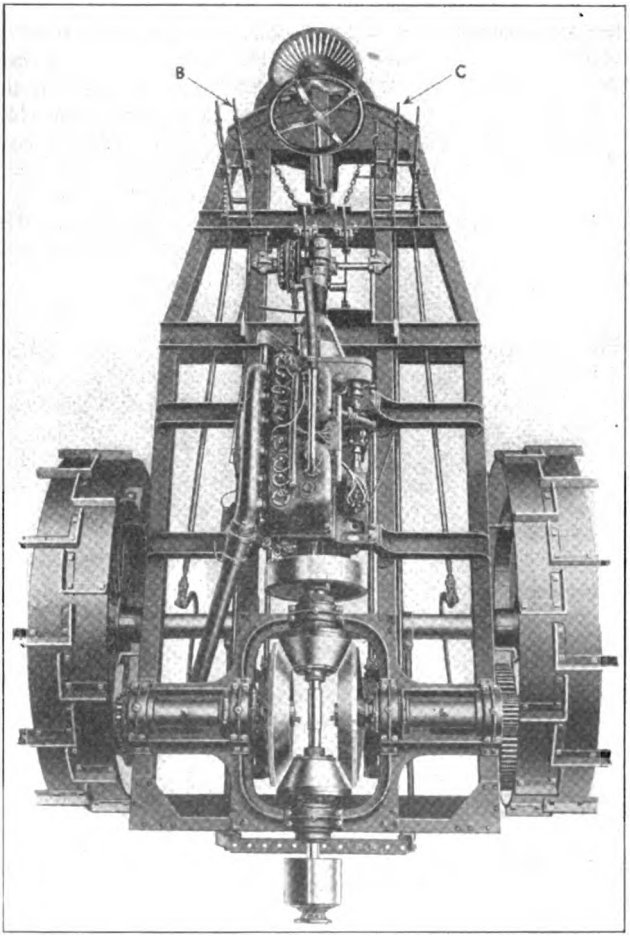

FIG. 4—FRICTION TRANSMISSION OF THE SQUARE TURN TRACTOR

the erecting floor; design ample bearings and shafts, arrange to protect the friction surfaces from dirt and oil, make friction members of sufficient size to require only reasonable pressure, and then let the user know at

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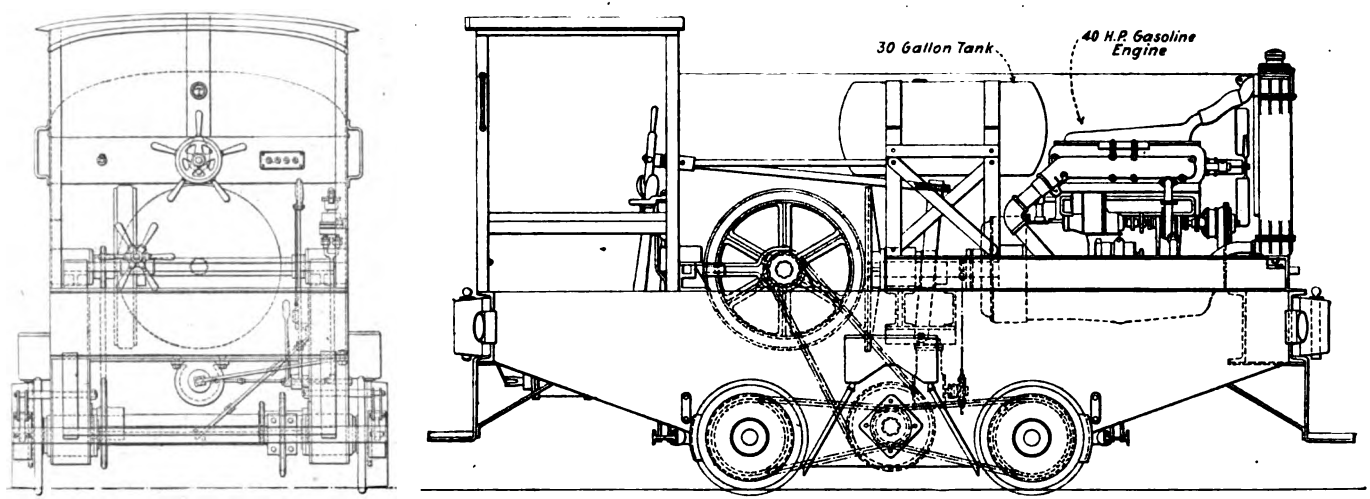

FIG. 5—CUMMINGS SIX-TON LOCOMOTIVE, END AND SIDE ELEVATIONS. THE FRICTION TRANSMISSION MEMBERS ARE BROUGHT TOGETHER BY SCREW PRESSURE

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least the basic principles of the drive which he is to operate. If this is done the drive will be successful, and will be just as dependable as any belt, chain and sprocket or toothed-gear drive. If the drive fails to handle its load or wears badly, do not blame the transmission, when the fault is due to inadequate size or to improper installation. There is a real advantage to the tractor in the extreme simplicity, the flexibility, the soft velvet clutching action, and the high efficiency of the friction transmission. It means longer life of the entire machine, greater ease in handling, and more work done per gallon of fuel.

Friction Transmission Types

Fig. 2 shows a simple form of disk friction drive mounted in a substantial frame, as applied to the Elgin tractor. Three pins projecting from the engine flywheel engage the spider that drives the disk shaft. The pressure is applied through the lever shown. Six forward positions and a reverse are provided, thus permitting the engine to work at its best speed regardless of the load. When used for stationary power purposes the pinion on the countershaft is disengaged by pulling the button shown above the left-hand bearing.

Fig. 3 illustrates the double disk transmission on the Heider tractor. In this arrangement the fiber friction member is a part of the flywheel; both disks are keyed to the countershaft, which slides endwise to bring the disks into frictional contact with the fiber, being moved to the right to go ahead and to the left to back. To increase speed the engine, and with it the fiber friction wheel, is moved toward the center of the disks. Seven speed positions are provided. This machine has been built for ten years and large numbers are in use.

The National tractor has a similar transmission arrangement.

The transmission in the Port Huron tractor is similar to that in the Heider, but instead of moving the engine when changing speeds the friction wheel slides on the rearward extension of the crankshaft.

In the Square Turn tractor, Fig. 4, only one speed reduction is provided. The levers, *B* and *C*, operate eccentrics carrying the large bevel friction wheels. The operator can cause both road wheels to go forward or backward, or one to go back and one forward at the same time. The machine has a single castor wheel at the narrow end; this is the rear end when plowing, the plows being underneath. When towing, the driver's seat is

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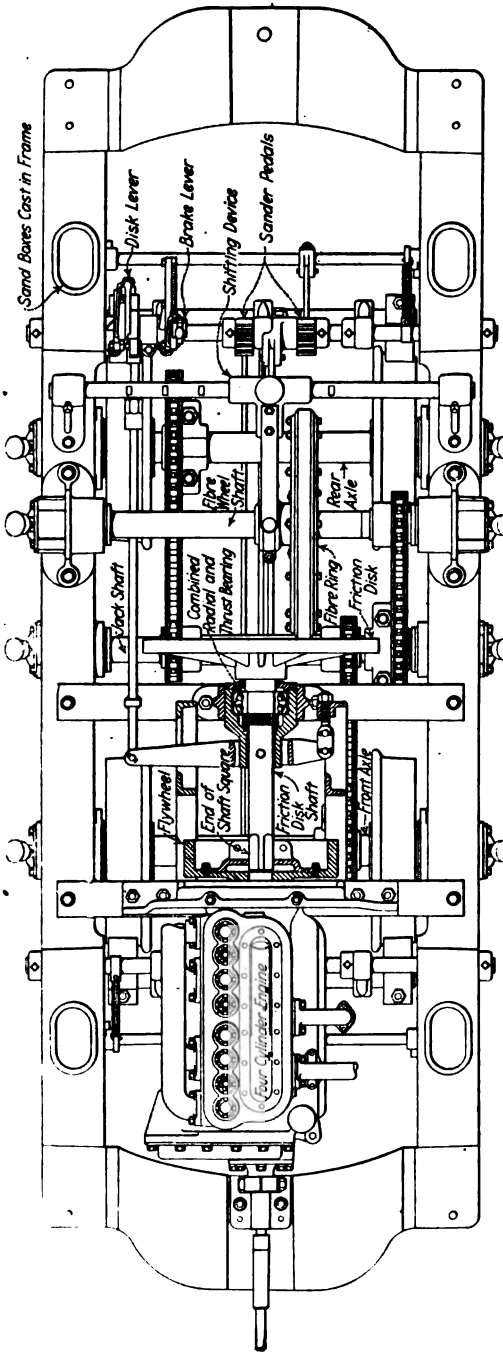

FIG. 6—PLYMOUTH LOCOMOTIVE, PLAN VIEW, SHOWING THE FRICTION DRIVE

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swung around to the opposite side of the steering-wheel, the driving wheels then being in the rear. No differential is required. The machine weighs 7800 lb., has 70-in. driving wheels and a 40-hp. engine.

Fig. 5 shows how the friction transmission members of the Cummings 6-ton locomotive are brought together by screw pressure, the screw being operated by a hand-wheel. The disk and friction wheels are 28 in. diameter. A number of these are now being built for the Government ordnance departments. The driver must be taught to lessen the pressure after getting his train under way or the fiber wheel will be short-lived. The speed change in this drive is obtained by the use of rack and pinion.

The builders of the Plymouth locomotive, Fig. 6, have been constructing friction-transmission equipped commercial and industrial vehicles for eleven or twelve years. On friction-driven locomotives it is the usual practice to arrange the friction wheel so that it will slide entirely across the face of the disk in order to give the same speed whether going forward or back (see Fig. 7), while on tractors it is usual to move the friction wheel past center only enough to obtain a satisfactory reverse speed, which is usually fairly slow.

A noiseless universal coupling, in which four flexible blades connect the driver on the end of the disk shaft to the flywheel is shown in Fig. 8. This device has been thoroughly tried out; it permits sufficient end-play so that the friction surfaces may be brought together or disengaged. It needs no center bearing, does not wear and is cheap. Fig. 9 shows a type of split filler or fiber band which is especially convenient for making repairs without disturbing other parts of the mechanism.

THE DISCUSSION

GEORGE A. WEIDELY (M. S. A. E.):—I noticed that on some of the tractors illustrated the fiber friction member was the driver. On others the fiber friction was driven. Just which is the proper method and why?

W. D. HAMERSTADT:—Theoretically I do not believe it makes much difference whether the fiber wheel drives or is driven. One practical condition that enters, however, is the wearing of flat places on the fiber. That can occur only when the fiber wheel is driven. Although such trouble constitutes one of the principal talking points against the friction drive, as used in automobiles, it does not occur as much as people may think. I have operated

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friction drive cars very extensively with the fiber driven member and never encountered such trouble.

Service Differs on Tractors and Automobiles

In tractor designs where there is a relatively great speed reduction through the friction members, the large diameter metal disk used as a driven member seems to work out best. I believe, however, the most important consideration is the nature of the work. Tractors operate by far the greater part of the time at low speeds and under the heaviest loads, as in plowing. Now, the highest efficiency in the variable-speed disk friction drive is when the fiber wheel is at the outermost point of the diameter of the disk. Therefore, to obtain highest efficiency during the period of greatest work and when most needed, as on heaviest pulls, I would recommend the fiber as the driving member. The condition is exactly reversed with automobiles; most of the service is on high speed with light loads. For such service I would recommend the fiber member be the driven member. In either case the load-carrying ability of the transmission should be at all times equal to that of the engine.

Slippage and Coefficient of Friction

D. L. GALLUP (M.S.A.E.):—What is the reason for the coefficient of friction of spur, bevel and miter gear drives being less than for variable-speed drives?

MR. HAMERSTADT:—The point has been brought out by Mr. Trask. Spur, miter and bevel drives have a motion of true rolling contact between their faces, and, like belt drives, they are designed for a uniform constant slip of say 2 per cent. The value of the coefficient of friction for such drives is, therefore, that value corresponding to 2 per cent slip on the slip-coefficient of friction curve (after deducting 40 per cent as a factor of safety for actual working conditions). In disk friction drives there is a certain twisting action of the fiber wheel against the face of the disk, and for each different position of the fiber wheel on the face this twisting action and the total average slip vary. Taking a case in which the slippage on the fiber is 4 per cent at a point near the center of the face of the disk, 3 per cent at a point midway between the center and outer edge of the fiber, and 2 per cent at the outer edge, each of these points will be pulling with a different value of the coefficient of friction corresponding to its rate of slip on the slip-coefficient of friction curve. As the load increases there will be a tendency for

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each point to slip more, so as to have a greater pulling capacity. A glance at the slip-coefficient of friction curve readily shows that the different points with their varying slippage exert maximum pull only as the slips approach values corresponding to maximum coefficient values at the peak of the curve, say 6 per cent slip. This average maximum value of the coefficient of friction is used, therefore, as the basis of design for variable-speed friction drives, of course, after allowing the 40 per cent factor of safety. It is not based on a predetermined fixed value of slip as for spur, miter and bevel drives.

MR. GALLUP:—I supposed that spur, bevel, and miter drives were more efficient.

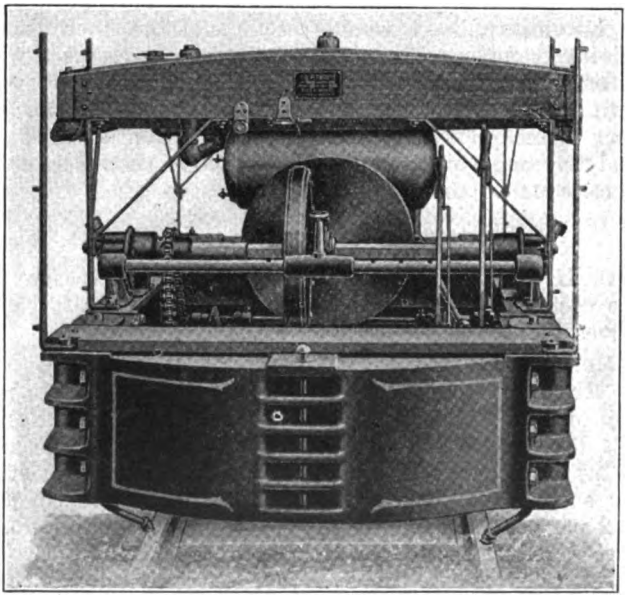

FIG. 7—PLYMOUTH LOCOMOTIVE, REAR VIEW, SHOWING PROVISION FOR SLIDING THE FRICTION WHEEL ACROSS THE FACE OF THE DISK

MR. HAMERSTADT:—They are more efficient than variable-speed drives. However, that statement may lead one to underestimate the efficiency of the variable-speed drive, whereas it is very good. The curve shows that the maximum value of the coefficient of friction is reached at about 6 per cent slip, that is, the maximum capacity is reached with a slip never exceeding 6 per cent. After that, the decreasing values of the coefficient of friction will simply

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result in stalling the drive. The 6 per cent slippage loss represents practically the entire loss in efficiency except the friction loss in shaft bearings and the slight loss caused by rolling friction of the two surfaces. In proper design the shaft bearings should cause little loss and with a dense hard smooth fiber surface the rolling friction seems almost negligible. We might say, therefore, that with friction surfaces clean and in good operating condition, as must hold true for the slip-coefficient of friction curve shown, the efficiency will never be less than 94 per cent at maximum load, and at lower loads it will be even greater. I do not believe there is a toothed-gear drive that can show continuously as high an efficiency.

C. S. CRAWFORD (M. S. A. E.):—We design toothed-gear drives to transmit power as efficiently as that but they are not manufactured that way.

MR. GALLUP:—What is the maximum commercial capacity, in horsepower, of a friction drive?

MR. TRASK:—I do not think it has ever been reached.

MR. GALLUP:—Two thousand horsepower?

MR. TRASK:—I see no reason why that figure cannot be reached. I do not know what is the largest drive we have turned out, but 5-in. shaft sizes are not uncommon.

The Panama Canal Commission has installed machinery for handling coal at Balboa and Colon where the storage capacity is, I believe, about 900,000 tons. A number of large bevel friction drives are used for handling this coal; most of the shafts, as I remember them, are about 5 in. diameter and they run at fairly high speed. I believe the units handle 3000 tons per hour.

Protection of the Friction Members

A MEMBER:—We have a tractor with a straight-faced disk. What provision is made against trouble from sand and grit?

MR. TRASK:—The friction members are not left exposed as shown in the illustrations. Generally they are protected just as carefully as toothed-gear transmissions.

A MEMBER:—Is there any difference in the capacity when such a drive is inclosed, particularly when grease or oil gets on it? How does lubricant affect it?

MR. TRASK:—Oil and grease must be kept off of the friction surfaces. Any mechanism operating close to the friction surfaces can be shielded in some way to prevent throwing lubricant.

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A MEMBER:—Should the surfaces be absolutely dry?

MR. TRASK:—They should be shielded from road splash and rain. However, we have found that a very small sprinkling of oil does not cause much harm.

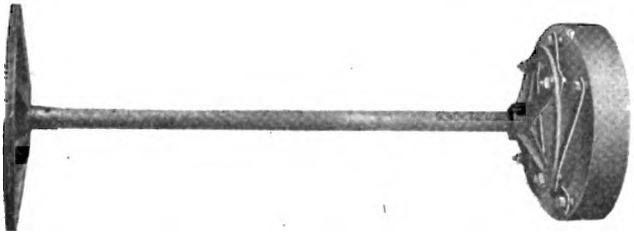

FIG. 8—NOISELESS UNIVERSAL COUPLING FOR FRICTION TRANSMISSIONS

MR. CRAWFORD:—Are compounds of any kind used occasionally on the disks, as for belt drives?

MR. TRASK:—We have experimented with dressings but it has been found that best results obtain from incorporating a small amount of compound in the fiber when it is being made.

Friction Wheels Operate Better in Damp Weather

MR. HAMERSTADT:—A fiber friction wheel invariably operates better in damp weather. It seems to pull more owing to the fact that the material absorbs moisture. Any friction dressing must, therefore, be hygroscopic in nature. The moisture absorbed seems to toughen the fiber. Rosin or greases should never be used. In the presence of heat due to slippage, the rosin will melt and then cause slippage. Oil and grease soak into the fiber, reduce the coefficient of friction and cause a high rate of slippage. After becoming absorbed it is almost impossible to get them out.

MR. CRAWFORD:—Have devices ever been applied to these tractors for truing the fiber member in case flat spots or anything of that kind should develop?

MR. HAMERSTADT:—That used to be brought up as advisable for automobile friction drives, but I never heard of it or the need of it for a tractor. I can see no real commercial need for that attachment on the machine. The friction rings are made with a split lap, and one of those wheels can be replaced in a half hour.

Comparison with Belt Drive

MR. WEIDELY:—This friction drive has been likened to a belt a number of times, but I do not see the simi-

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larity. A friction drive depends upon certain contact pressure, and Mr. Trask made the statement that it did not require any considerable area of contact. A tight belt is not required in a belt drive, but smooth surfaces and many square inches of belt contact are needed. According to Mr. Trask these are not needed in a friction drive.

MR. HAMERSTADT:—The belt drive is similar to the Gabriel snubber. An enormous pressure is introduced against a body by taking one or more wraps around it with a cord or band and then pulling on the cord. We all know the effectiveness of the cable drum or winch. Area absolutely is not required in a belt drive to transmit power, but we do need the wrapping pressure of the belt against the pulley face just as in the Gabriel snubber. A belt one inch wide will transmit the same power to a pulley as one ten inches wide if they both have the same arc of wrap, the same tension and the same speed.

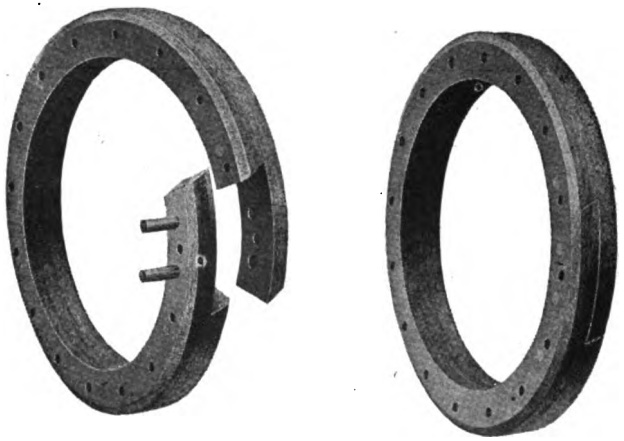

FIG. 9—SHOWING CONSTRUCTION OF A SPLIT FILLER OF FIBER BAND FOR A FRICTION TRANSMISSION

A MEMBER:—Then why is compound used on a belt?

MR. HAMERSTADT:—A compound or dressing applied to a belt will increase the grip or value of the coefficient of friction between the surfaces. Of course, if we can increase the value of the coefficient of friction, then for a given pull the pressure need not be so great, and likewise the tension in the belt that causes the pressure may be less; in other words, belt dressing will permit a slacker belt, as long as its effect lasts.

MR. CRAWFORD:—The most satisfactory belt drive,

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however, is with a slack belt having a large arc of contact on the pulley.

MR. HAMERSTADT:—Yes; if the arc of wrap of the belt is great, it results in a much greater total pressure of the belt on the pulley, and, of course, the power that can be handled is dependent on this pressure.

Friction Drive Rim Speed

A MEMBER:—Is there a definite speed at which the friction drive members should operate? Will it give extra efficiency or under efficiency or burn itself up by going too fast?

MR. HAMERSTADT:—I know of no particular limit. Friction surfaces can and do commonly operate at speeds of 6000 to 7000 ft. per min. Of course, for a given amount of power, the higher the rim speed the less the tangential pull, and therefore less pressure is required between the friction members. Like the belt drive, it is economy to operate at high speeds. I have known of frictions to operate at speeds as great as 13,000 ft. per min., but this seems too high.

In any case, as the speed increases, the pressure on the friction surfaces should be reduced, as otherwise the "flexing" or "working" of the fiber at the high speeds will generate excessive heat and do damage.

GEORGE T. BRIGGS (A. S. A. E.):—Should there be a definite size depending on the amount of load? Are there varying sizes of fiber wheels?

MR. TRASK:—There are no fixed stock sizes of friction wheels. Any size within the limits of mechanical construction can be supplied and by following formulas which I have given any engineer can tell what size to use.

A MEMBER:—If it is desired with a belt drive, to double the power transmitted at a given speed by two members (one driving and one driven), the width of the belt could be increased. What is done with a friction drive for a similar case? Is the pressure or the width of the driving disk increased or what is the limiting factor?

MR. TRASK:—The surrounding conditions govern that. If space were available they might be increased in diameter. In a belt drive the diameter of pulley or width of belt can be changed. In a friction drive the diameter of friction wheel or its width of face can be increased.

MR. HAMERSTADT:—Let me point out in this connection that to increase the width of belt or the face of the

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friction members does not in either case, in itself, result in greater transmitting capacity. Belt widths are increased so that the total section of the belt will be sufficient to stand the tension under which it must operate, without too great a stretch; in other words, to increase the thickness of a belt will have just as much effect as to increase its width. Likewise, the face widths of friction members are increased only to give a proper distribution of the total pressure necessary to handle the load. In all cases it should be remembered that it is not safe to apply more than a certain fixed pressure against each one-inch width of face and greater total pressures can be safely used only by increasing the face widths. Pressure in friction drives and belt tension in belt drives are really the important limiting conditions of these forms of transmission. Use of excessive pressures will invariably result in rapid breaking down and destruction of the fiber.

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METROPOLITAN SECTION PAPERS

ELECTRICALLY-CONTROLLED STEAM AUTOMOBILES

By Abner Doble*

The first actual attempt at driving a vehicle by steam seems to have been made in 1770 by Cugnot. A single wheel in front served as a steering and driving wheel. A pair of cylinders acted upon a crankshaft which was geared to the rear axle. The boiler was located in front of the engine, and both boiler and engine overhung the front wheel. This vehicle was capable of making a speed of 2¼ m.p.h., but was unmanageable because of steering difficulties. The original model is preserved in the Conservatoire Des Arts et Metiers in Paris.

The first authentic record of a power-propelled vehicle is found in a patent taken out by Watt in 1784, which describes the application of a steam engine for the purpose of locomotion.

Speedily following this, in 1786, a steam carriage was built by Murdock, Watt's assistant, and was run upon the high road near Redruth, Cornwall.

Similar experiments were made by William Symington, to whom the idea probably occurred at about the same time as to Watt.

Oliver Evans, an American inventor, likewise constructed a steam vehicle in 1787, and obtained rights to operate steam vehicles in Pennsylvania and Maryland.

Some years later (1802) other attempts were made by Trevithick and Vivian in England, but were shortly discontinued, probably owing to the generally poor condition of the roads, or to the attraction of enterprise and capital to the improvement of railroads, the introduction of which had been begun.

For about thirty years the subject was in abeyance, when it was revived again by Griffiths, Brunel, Gurney and Hancock. The steam carriages constructed by these inventors were run in various parts of England and Scotland, but their operation did not prove profitable commercial undertakings. In 1839 Hancock built and operated several steam omnibuses in the city of London and the suburbs.

*Vice-President, Doble-Detroit Steam Motors Company

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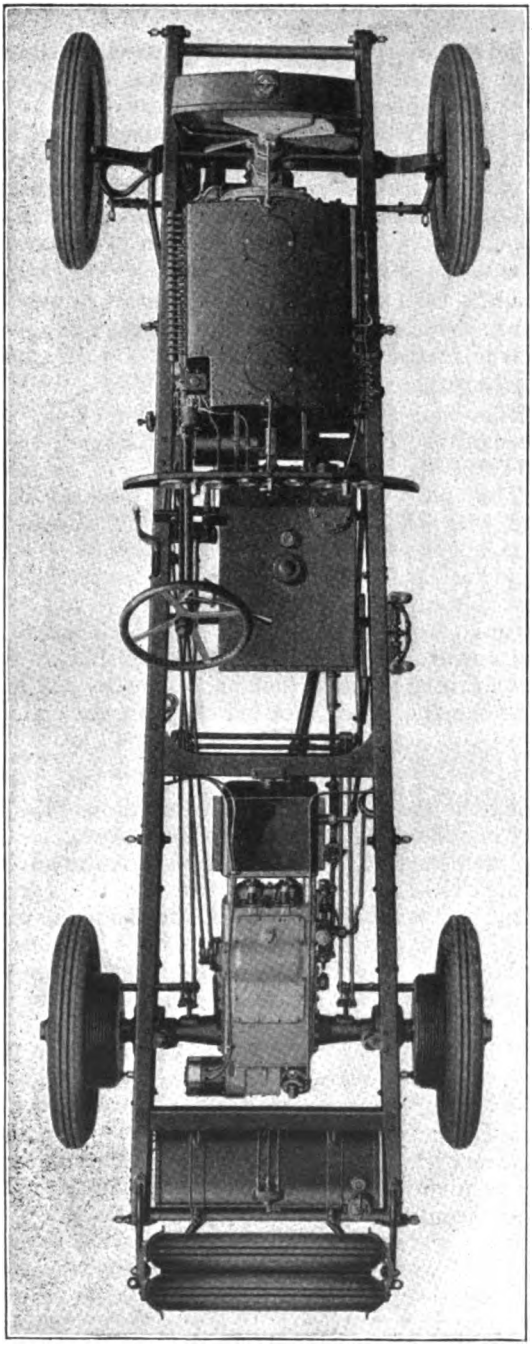

CHASSIS OF THE DOBLE STEAM AUTOMOBILE SHOWING POSITION OF POWERPLANT UNITS

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The latter type of vehicle, when carrying supplies of water and fuel, but without passengers, weighed about 3½ tons. In general it attained a speed of 10 m.p.h., but on exceptionally good roads was capable of a speed of about 20 m.p.h. However, prohibitive tolls were extorted on the turnpike roads and road laws passed which provided that a man had to walk in front of a car waving a red flag, so that finally the operation of steam vehicles was practically stopped.

Not until 1889 was there a substantial revival in motor-driven vehicles, and from that time on rapid strides were made in the development of the so-called horseless carriages.

Serpollet had been experimenting for some years in France, and produced some efficient machines, the boiler being of the flash type, using gasoline or kerosene for fuel. The fuel was first vaporized, by means of a cup charged with alcohol placed beneath the burner, which served to heat the lower part of the coil. The vapor from the fuel in the heated coil passed to the base of the burner, where air was introduced with the vapor, the resulting mixture being ignited by the preheated coil in passing upward. The coil was then kept hot by the continuous burning of the fuel and likewise continued to vaporize the fuel as it was fed through the coil.

Early Steam Car Builders

Among the early builders of steam automobiles in this country may be mentioned the Baldwin Automobile Company, Providence, R. I.; the Milwaukee Automobile Company, Messrs. F. E. & F. O. Stanley, the Locomobile Company, and a little later the White Company of Cleveland.

In the early nineties a wave of progress spread throughout the Continent, England and the United States, which started the development of the internal-combustion engine, as well as the electric motor for road vehicles.

In the steam car several great disadvantages had presented themselves. One was the low mileage obtained from the amount of water that could be carried conveniently. It was most uncommon to be able to make a continuous run of one hundred miles without refilling the water tank, and in general, it was necessary to stop for water every twenty-five to fifty miles of travel. Another serious drawback was the trouble and time required in starting. If a steam car could be got

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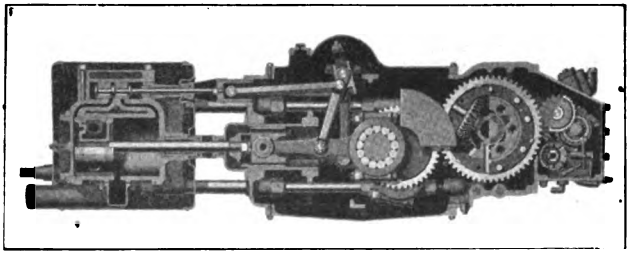
under way in ten to fifteen minutes from a cold boiler, it was considered most satisfactory. Ordinarily it might require a half hour and the manipulation of various manual devices, which demanded an intimate knowledge of the mechanism of the car. A great deal of the delay in starting was due to the preheating of the main burner by means of a pilot light, which first vaporized the fuel in the primary burner. There has also always been an element of danger from the exposed flame of a pilot light when gasoline is used.

About ten years ago steam cars had reached a high state of popularity and had a large number of enthusiastic advocates, since two companies—the Stanley and the White—were producing them in quantities. Under these



CROSS-SECTION OF DOBLE ENGINE
Generator gear meshes with main drive gear on rear axle

conditions and in an industry which was bound to assume such huge proportions, it was most natural that experiments should be made with other sources of power besides steam.

At the time of this early development in automobiles even the long established prestige of the stationary steam engine seemed to be somewhat in jeopardy, and a number of large engine builders began to think of the possibility of steam being replaced by the explosive type of engine. A great deal of attention was being given to the perfection of the gas engine for stationary plants. In such service the problem of the ignition of fuel and the generation of steam did not enter as a factor, as it does in the application of a steam generating plant for an automobile.

It was most natural, therefore, that automobile engineers should and did turn their attention almost entirely to the perfection of the internal-combustion engine. Development along this line progressed most satisfactorily, one by one the manufacturers of steam

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cars changed over, and with the exception of those of the Stanley Company, no steam automobiles have been in regular production for the past six years.

Every engineer will agree that steam is the most dependable and the most flexible source of power. To approach that smooth running, positive power in a gasoline car, the tendency has been to increase the number of cylinders, until we now have in common use many eight and twelve-cylinder internal-combustion engines. Other methods have also been employed, such as increasing the number of valves and combining the gasoline engine with the electric motor. All of these practices tend to complicate the mechanism and to increase the fuel consumption in the effort to secure the performance obtainable with the steam engine.

During my ten years of experimental work I have always had in mind the possibility of overcoming the objectionable features and seemingly insurmountable difficulties that had diverted development from steam to internal-combustion engines for the propulsion of automobiles. The principal problems that confronted me were:

1 The low mileage possible on one tankful of water.

2 The formation of scale in the boiler with a resultant drop in efficiency and added liability of burning the already extremely hot heating surface, which in turn necessitated the frequent cleaning of the boiler.

3 The toil, time and inconvenience involved in firing up the main burner by means of a pilot light.

If the ignition and starting of a gasoline car could be accomplished almost instantly by the operator sitting at the steering wheel, why could not the same result be obtained in a steam car?

Electricity is a refinement now used for the ignition and starting of the steam car. The driver can sit at the steering wheel, close the switch on the steering column, and start his car in a minute's time, even though the car may have been standing idle for several hours.

At extremely low temperatures, with the generating system cold, it will require about two to three minutes to start the car, but from the time it begins to move, smooth, even power is had, and there is not from five to ten minutes of bad running as is the case, until the engine warms up, in a gasoline car.

One filling of the water tank, holding 25 gal., will suffice under ordinary conditions for upwards of 2500 miles of running. One filling of the 25 gal. kerosene tank insures from 250 to 260 miles of travel.

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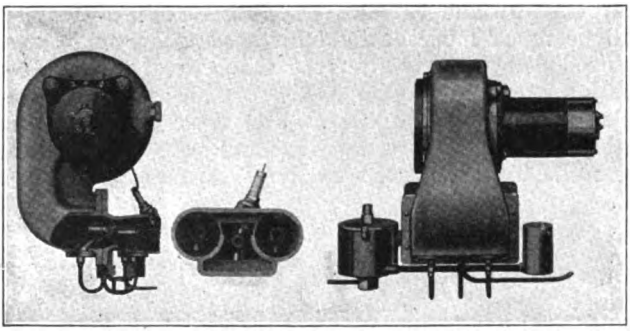
The simplicity of construction and operation of the present-day steam car makes the predominant features of electrically-controlled steam most apparent. The automatic controls are so simple in construction and so positive in action that almost no attention need be given to them.

The moving parts in the steam-car mechanism are about one-half the number required in a six-cylinder gasoline car. That means less worry and less expense for upkeep.



BLOWER, MOTOR AND CROSS-SECTION OF FUEL NOZZLES
The starting nozzle is forward and the two running nozzles are side by side slightly behind the starting nozzle

These accomplishments have been the result of exhaustive experiments to determine the most efficient and most adaptable type of boiler to use. I was able to profit by the experience of others who had already done a great deal of good work toward perfecting the application of the flash and fire-tube types of boilers.

Six years were spent in building and testing experimental steam powerplants, which resulted, late in 1913, in trying the combination of a water-level type of boiler with a honeycomb radiator to condense the exhaust steam. The theory was that in order to obtain the necessary cooling surface this type of radiator furnished the best possibilities for obtaining an adequate distance upon one supply of water. The results were most gratifying. The car would run from 1000 to 1500 miles on 24 gal. of water.

The honeycomb radiator furnishes about six times the cooling surface of any other type of equal size, but the selection of this type had previously been deemed unwise because the heavy molasses-like oil used in steam

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engines clogged up the extremely small radiator passages, and the exhaust steam, particularly when a flash boiler was used, was liable to melt the soldered joints of the radiator. It was my belief that the use of a heavy oil was not essential.

In a steam automobile little lubrication is required for the cylinders, as the piston speed is low at ordinary driving speeds, and the surface of the cylinder, which is of cast iron, is easy to lubricate.

The presence of moisture in the steam during expansion goes a long way toward the proper lubrication of the cylinders and valves. The cylinders in a steam car will be much more easily lubricated when moist steam is used for the driving force than in an internal-combustion engine when hot gases furnish an explosive impulse within the cylinders. For these reasons it was determined to try ordinary gas-engine cylinder oil, and from the first it proved entirely satisfactory.

Development of a Steam Generator

Attention was next given to the development of a satisfactory steam generator. The water-level type of boiler, which had been used successfully in conjunction with an improved condensing system, possesses a number of the essential qualities required; it holds the temperature of the steam practically constant with no danger of exhaust temperatures sufficiently high to melt soldered joints or effect an undesirable change in the lubricating oil. It always keeps a large reserve of water heated to the steam temperature, which gives it steaming stability and admits of great acceleration. The heat-transference conditions make for efficiency, owing to the extremely short distance through which the gases of combustion radiate their heat to the tube walls.

On the other hand, the fire-tube boiler which we first tried had its disadvantages; it was heavy and costly to manufacture, since it had to be wound with a mile of piano wire to minimize the potential danger present in a large-diameter shell. It was liable to leaks, which might be caused even by overheating with low water or by oil working through the expanded joints where the tubes were fastened into the heads.

The flash type of boiler was out of the question, but it had certain good qualities. The direction of the water flow opposite to the flow of the gases of combustion was a great advantage, in that it allowed the water to ex-

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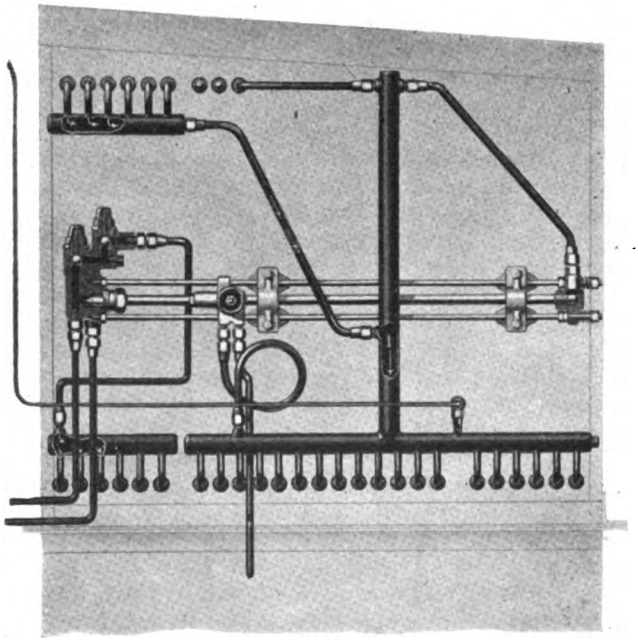
tract, so far as possible, the last remaining thermal units.

The long period of experimentation has resulted in our present steam generating system, which is similar in theory to the flash boiler, yet in appearance is more



WATER LEVEL REGULATOR

When tube is filled with water, needle-valve opens owing to contraction of tube; water from pumps is by-passed to water tank

like the water-tube boiler, having a water level in the evaporating zone. It consists of twenty-six sections placed in a planished steel casing insulated with kieselguhr. Each section is made of cold drawn seamless steel tubing of ¾-in. diameter, No. 16 gage metal. Twenty of these sections are used for the generation of steam, and the remaining six are the preheating, or economizer sections. The generating sections consist of twenty vertical tubes connected top and bottom by horizontal headers. The vertical tubes are swaged at both ends to about three-eighths of an inch, and are welded into the horizontal headers by the autogenous acetylene process, thereby making the section in effect one piece of steel and actually stronger at the welded joints than in the tubing itself. The six economizer sections are like

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the generating sections, except that they have sixteen vertical tubes in place of twenty.

The generator is designed for a working pressure of 600 lb., and the safety valve is set for 1000 lb.; each section is tested to withstand 5000 lb. pressure. When sections have been subjected to a destructive test in laboratory experiments, observations have shown that rupture of a tube occurs only at cold water pressures ranging from 8000 to 9500 lb., and that when a rupture does occur it is invariably at a point remote from the welds.

Directly below the generating sections is the combustion chamber, and below the economizer is the exhaust for the gases of combustion. A bridge wall of kieselguhr divides the two sets of sections. A manifold, through which water enters, connects the lower headers of the economizer sections. The water leaves by a similar manifold at the top and is led to a manifold connecting the lower headers of the evaporating sections. The steam leaves the upper headers and is conducted through a fourth manifold to the throttle valve.

Besides being absolutely free from any danger of explosion a boiler of this construction can be manufactured cheaply, and any damaged section can either be easily and cheaply replaced, or be isolated in a few minutes by blanking it off until it can be replaced. The excellent heat transference conditions due to the close and regular heating surfaces virtually duplicate those of a fire-tube boiler, while a large reserve of water close to steam temperature is always present. The flow of the water counter to that of the gases, with no circulatory flow, and the all-steel construction show a distinct similarity to the flash type. Water is supplied to the boiler by a crank-driven pump, and the water level maintained about half way up the generator by an automatic regulator. If the water level falls below normal the regulator tube will fill with steam, the expansion of which closes a by-pass valve, thereby allowing water from the pump to enter the boiler. As soon as the level reaches normal, the regulator tube fills, through an outside pipe, with water which has not been in circulation in the generator and is therefore comparatively cool. The regulator tube at once contracts, permitting the valve to open and all water to be by-passed back to the supply tank.

We now come to the two most important factors in the perfection of operation of a steam automobile:

1 The ignition system, which permits the driver, seated at the steering wheel, to cause the lighting of the fire in the boiler by the turning of an electric switch.

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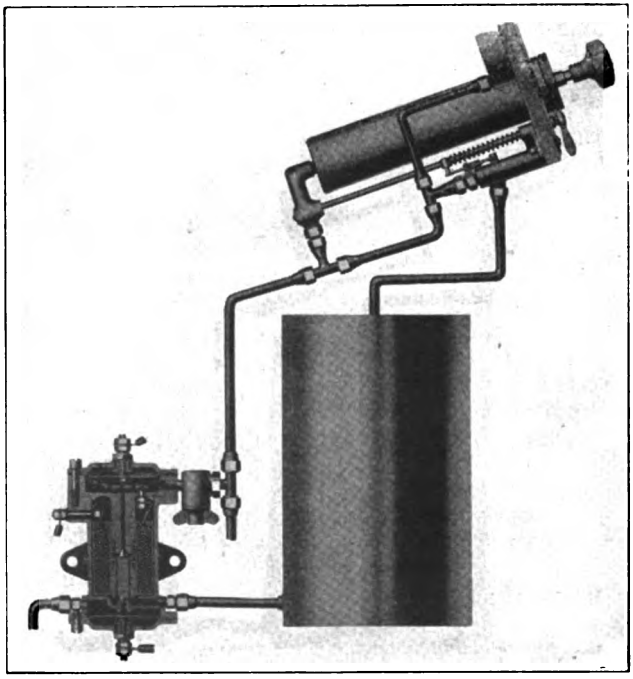
2 The small combustion chamber, in which enough fuel can be consumed to supply the requisite number of heat units to generate and maintain a sufficient steam pressure at all times for driving the engine under whatever road conditions may be met.

Ignition System

In order to start the car, it is necessary to have 4 lb. air pressure on the fuel tank; kerosene is then delivered to the float chamber, which is located alongside the spray nozzles in the venturi tubes of the blower. This 4-lb. air pressure is obtained first by means of a hand pump on the instrument board, but when the car is running, it is maintained by a small air-pressure pump operated by a cam on the rear-axle shaft. The storage battery is charged by a generator driven through a gear on the axle shaft.

When the switch on the steering column is closed, the current passing through a solenoid starting valve starts the blower motor. The blower, revolving at about 2300



Solenoid Starting Valve with Auxiliary Air Tank

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r.p.m., creates a forced draft which by aspiration feeds the kerosene from a float chamber with a proper mixture of air through what are termed the two running nozzles. At the same time that the blower is started by the current passing through the solenoid, the latter magnetizes and raises a small valve which releases air under 4-lb. pressure from a small auxiliary air tank connected with the solenoid valve. This charge of air rushes to a starting nozzle located between the two running nozzles, aspirating kerosene through the starting nozzle with a rich mixture of air.

Directly in front of the starting nozzle is a spark-plug. At the time the air pressure is released through the solenoid valve, it acts on a diaphragm in the solenoid and makes an electric contact with a spark coil so that simultaneously as air and kerosene are passed through the starting nozzle in an atomized condition, the spark ignites the rich mixture, and this primary flame ignites the lean mixture which is being fed by the blower through the running nozzles.

The duration of the primary flame at the starting nozzle and the duration of the spark are governed by the time required to exhaust the air supply in the auxiliary air tank at the solenoid. This is about one and one-half seconds.

But the motor is still driving the blower so that the fuel through the running nozzles which has been ignited by the flame of the starting nozzle continues to burn in the combustion chamber.

By a simple automatic regulator, the current to the motor is shut off when the steam pressure has reached 600 lb., and the fire goes out in the combustion chamber. At the same time, the auxiliary air tank is recharged by the solenoid valve dropping to its seat due to the breaking of the current, so that the system is ready to repeat the starting operation which is now being controlled by the steam pressure regulator. When the steam pressure drops to about 550 lb., contact through the solenoid is made again, and the fire is lighted. This requires no attention on the part of the driver and the fire continues to go on and off according to the steam pressure, so long as the switch on the steering column is closed. Driving over ordinarily level roads, the fire will be on about one-third of the time.

Many difficulties presented themselves in working out the details of this method of ignition. At the outset it was demonstrated that if a mixture of sprayed kero-

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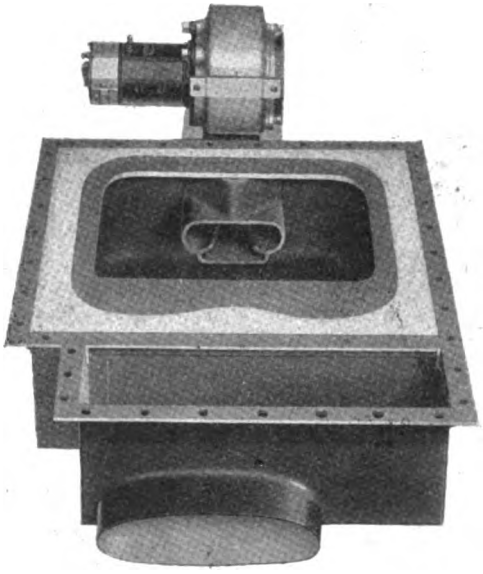
sene and air was formed into which a sufficient amount of air was injected to insure a complete combustion of the mixture, it could not be ignited by the small flame produced by an electric spark.

In order that the entire system respond to the closing of a switch on the steering column it was essential, of course, that ignition of the fuel should take place from an electric spark. It was discovered that a flame of considerable proportion was necessary to ignite the



COMBUSTION CHAMBER SHOWING THE VENTURI TUBES
Starting nozzle in center tube, two running nozzles in outer tubes

lean, rarefied mixture necessary for complete combustion. After finding that this could not be ignited by an electric spark, it was necessary to invent some other device for accomplishing the purpose. Experiments then led to the discovery that if a mixture of sprayed kerosene and air was formed in which there was an excess amount of liquid fuel, this rich mixture could be ignited by an electric spark. Therefore, it was arranged to use three spray nozzles. The center or starting nozzle was placed between the two running nozzles and slightly ahead of them, so that as the rich mixture in the starting nozzle was ignited the lean mixture being fed past the flame of the starting nozzle at high velocity would become ignited and continue to burn in the combustion chamber.

Of the many problems met in securing the necessary

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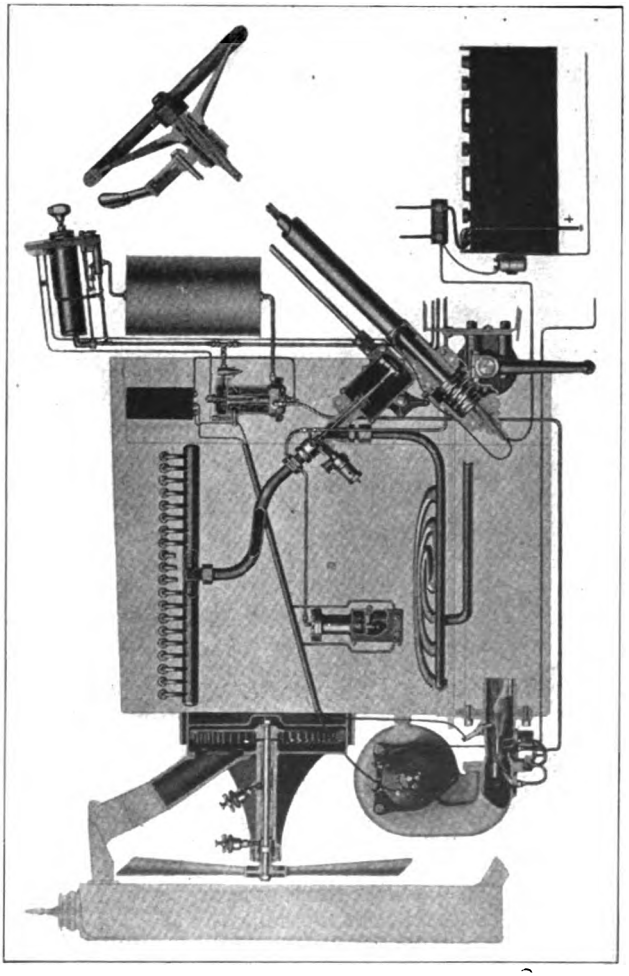
conditions, the most important was to avoid vaporizing or gasifying the liquid fuel used in the combustion chamber for the generation of steam. In all former attempts to burn liquid fuel, the vaporizing process was employed; in this a gas flame was produced which



AUTOMATIC DEVICES ON THE LIVE STEAM SIDE OF THE BOILER
Showing solenoid valve and pressure regulator which lights and shuts off fire as steam pressure varies between 600 and 550 lb.

could readily and easily be confined in the limited space available for the fire-box of an automobile boiler. To convert kerosene fuel into a gas necessitated the use of pilot burners or preheating devices that required considerable time to acquire a sufficient temperature before the gas flame was available. With devices of this na-

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ture, it has not been possible to construct a steam car in which the operator can sit at the steering wheel, close a switch and move off with that degree of promptness that has been made possible in the most highly developed gasoline cars.

The space available for the combustion chamber or firebox, of a steam boiler is limited in a steam-driven automobile—so limited, in fact, that without changing the conventional lines of the modern automobile hoods, a fire-box 20 in. square and 8 in. deep was as large as could be employed.

It is well known that if the flames from burning fuel be permitted in contact with the flues or water tubes of a boiler, the products of combustion will readily form smoke and soot, and that the tubes will soon become so heavily coated as to reduce materially the evaporating capacity of the boiler.

When oxygen in the form of air is supplied with the atomized liquid fuel, sufficient not only to support complete combustion, but also to produce a temperature high enough to insure ample steam pressure under all possible demands, an immense body of flame results.

The main problem, therefore, was to construct a fire-box of such shape that the sprayed fuel and air could be injected into it and the flaming combustible be retained within the limits of the combustion chamber and yet not come in direct contact with the water tubes of the boiler.

The method of burning the fuel consists in projecting a stream of sprayed kerosene and air in sufficient proportion to support complete combustion and with such force as to compel the stream to remain in a horizontal plane. The stream is doubled upon itself in a horizontal plane to confine it in parallel planes until combustion is complete.

This has been accomplished by arranging the shape of the combustion chamber so that the fire is almost completely confined in the fire-box and does not come into direct contact with the boiler tubes.

Construction of the Engine

The engine is exceedingly simple and contains but 11 moving units. The dimensions of all the working parts are ample to insure uninterrupted service under maximum conditions of load. The engine is 5-in. bore, 4-in. stroke, of the two-cylinder, single expansion, double action type. The unaflow principle is employed in order to provide the high expansion desirable, with a noiseless

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valve gear and only one valve per cylinder. The valve takes care of the steam inlet, while the exhaust passes out through ports uncovered by the piston at the end of the stroke. It is thus possible to secure cut-off at 3/16 of the stroke. This cut-off is used for all ordinary running, and operates with perfect accuracy on either stroke of the piston. For starting or heavy pulling a cut-off is used at three-fourths of the stroke.

Since the thermal conditions in the unaflow cylinder are practically ideal, it is unnecessary to use superheated steam, but it is desirable because it reduces the weight which must be condensed. The slide valves are on top of the cylinder, and are each made in two pieces so that in slow running they can be lifted whenever the compression exceeds the steam-chest pressure. This makes the engine run smoothly at all speeds, and also allows a high compression at higher speeds and steam-chest pressures.

The valves are actuated by a specially designed valve gear, which dispenses with the need for eccentrics, thus making a one-piece crankshaft possible. This gear gives an excellent steam distribution, and also reverses the engine without additional devices. It is a simplified form of the Joy valve gear, from which the correcting and anchor links have been eliminated. The rocker guide is straight instead of curved.

The piston rod passes through a specially designed gland, which is made in such a manner that no steam can blow by it. On account of the long bearing surface there is practically no wear, and repacking is rarely required.

The crankcase is an aluminum casting, and contains the moving parts of the engine, except the pistons and valves. The differential is also contained in the crankcase, and the taper-tubes of the axle bolt directly to it, thus making the engine and rear axle one unit.

The forward end of the engine is suspended from a cross-member of the frame by a flexible steel strap.

The crankshaft, differential and big-end connecting-rod bearings are annular roller and are of such proportions that no wear can occur during the natural life of the car. All of the other bearings, such as the wrist-pin and valve-gear bearings, are hardened steel, running in hardened steel bushings.

The power is transmitted to the rear axle by means of two large spur gears, a 42-tooth gear on the engine crankshaft and a 54-tooth gear on the differential. The

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engine has more power than is needed to spin the wheels on dry pavement, while the car is held stationary.

Method of Lubrication

Gas engine cylinder oil is supplied to the crankcase of the engine. This is splashed by the crankshaft and main driving gears and lubricates the working parts of the engine. By means of a small oil feed-pump, the oil from the crankcase is also delivered to the steam chest of the engine, lubricating the valve seats, valve stems and cylinders. This lubricator is set to feed one gallon for each 5000 miles.

Passing through with the exhaust steam from the engine a small quantity of oil continuously finds its way to the water supply tank and is fed with the water into the boiler. The oil is accordingly regularly pumped into the boiler along with the water, and far from having a deleterious effect, really performs its most valuable functions in that part of the powerplant. This oil is very thin at 490 deg. fahr., the approximate steam temperature at 600 lb. pressure, and the coating of oil which forms over the entire inner surface of the boiler, is consequently so thin as to have a negligible effect upon the heat transference.

As scale does not adhere to a surface coated with oil, the interior of the boiler remains entirely free from incrustations of scale matter, and is likewise thoroughly protected from corrosion. The second function of the oil is to coat each particle of scale-forming material as it is thrown out of solution, thus preventing one particle from sticking to another in such a way as to form a body of sufficient size to clog some restricted passage. No large amount of scale-forming material is introduced into the system since but little make-up water is required. The violent ebullition and constant flow of the medium toward the steam outlet causes the minute particles of scale to be carried along with the steam, so that the boiler and radiator are both kept free from deposits. The scale-forming material finally reaches the water tank, where it either remains or continues to circulate through the system without any apparent deleterious effect.

I have carefully examined the boiler and radiator of a car driven over forty thousand miles, and they were as clean as or cleaner than when the car was built. I do not believe that there could be more convincing proof of the entire effectiveness of the system.

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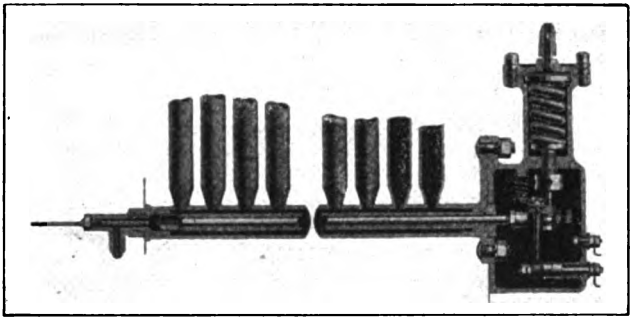
Utilization of the Exhaust Steam

The exhaust steam is carried from the engine to a small, low-pressure steam turbine which is mounted on the front of the boiler casing. The condenser fan is on the shaft of the turbine. By the use of the turbine, a greatly increased efficiency in condensation is effected, as, when the engine is working the hardest on grades or heavy roads, the turbine will be driven at an increased speed, so that the fan will draw more air through the condenser at the time it is most needed, regardless of the car speed.



Sectional View of Steam Pressure Regulator and Low Water Cut-Out

Advantages of Steam Power

Let us now sum up the advantages of steam for driving automobiles, as compared with the internal combustion engine. These are:

1 Torque range of 100 per cent with a maximum torque available at zero speed; change-gear mechanisms and clutch therefore unnecessary.

2 Mean effective pressure (and equivalent drawbar pull) always under control of the operator; varied by throttle and cut-off from zero to maximum, the maximum limited only by the tractive capacity of the rear wheels.

3 Utmost mechanical simplicity with a small number of moving units.

4 Low manufacturing cost, owing to simplicity of construction and lack of "fussy" work in production.

5 Entire absence of lubrication troubles; no contamination of crankcase oil by kerosene, gasoline, water, road dust or carbon.

6 Low fuel cost per mile.

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Performance in Actual Practice

The following tests were made without a superheating coil. It is my opinion that on our present boilers with superheaters, we are getting at least 20 per cent greater evaporation without priming than was possible on the earlier boilers with which these data were taken.

Total tube surface in boiler, sq. ft.	143
Total tube surface in economizer, sq. ft.	34
Total tube surface in boiler and economizer, sq. ft.	177
Total water surface in boiler, sq. ft.	118
Total water surface in economizer, sq. ft.	25.9
Total water surface in boiler and economizer, sq. ft.	143.9
Total capacity of boiler, cu. ft.	1.425
Total capacity of economizer, cu. ft.	0.333
Total capacity of boiler and economizer, cu. ft.	1.758
Total capacity of boiler and economizer, gal.	10.8
Total capacity of boiler and economizer, with 11 in. water level, gal.	9.
Total weight of boiler and economizer sections, lb.	450
Actual evaporation, lb. per hr.	630
Equivalent evaporation, lb. per hr.	765
Equivalent evaporation, per lb. of kerosene, lb.	18.05
Equivalent evaporation, per sq. ft. of boiler surface, lb.	7.38
Equivalent evaporation, per sq. ft. boiler and economizer, lb.	5.20
Fuel consumption, lb. per min.	0.687
Temperature of steam at boiler, deg. fahr.	515
Superheat, deg. fahr.	25
Pressure, lb. gage	600
Heating surface including economizer, sq. ft.	146.7
Overall efficiency, per cent	90
Temperature feed water, deg. fahr.	62
Boiler horsepower	22.25
Temperature flue gas, deg. fahr.	410

Water Consumption at 60 m.p.h.

Volume of cylinder = 19.6 × 4 = 78.4 cu. in.
Volume at 3/16 cut-off. 3/16 × 78.4 = 14.65 cu. in.
Volume per revolution for two cylinders 58.6 cu. in.
Volume at 60 m.p.h. or 740 r.p.m. = 740 × 58.6 = 43,400 cu. in. = 25.2 cu. ft.
Steam chest pressure, 250 lb.
Weight 1 cu. ft. steam @ 250 lb. pressure, 0.57 lb.
25.2 × 0.57 = 14.3 lb. water.

Engine Data

Bore, in.	5
Stroke, in.	4
Unaflow, double-acting, two-cylinder, Doble valve gear cut-off	3/16; ⅜; ⅝
Gear ratio to rear axle	42 to 54

General Data

Wheels, in.	35
Tires, in.	5
Wheels, r.p.m.	576
Engine, r.p.m.	740
Ratio dynamo speed to rear axle speed	9.43 to 1
Ratio dynamo speed to engine speed	7.33 to 1
R.p.m. at 30 m.p.h.—Wheels	288
R.p.m. at 30 m.p.h.—Engine	370
R.p.m. at 30 m.p.h.—Dynamo	2,712

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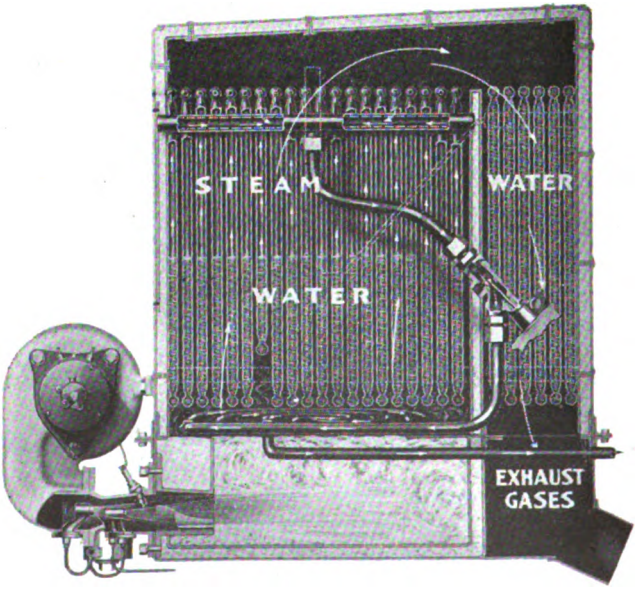
Manufacturing Advantages

The following manufacturing advantages of the steam car are greatly in its favor:

1 The much smaller amount of material required.
2 The small number and simplicity of machining operations.
3 Minimum investment in factory and equipment in proportion to quantity produced.
4 Comparatively small number of parts entering into the construction of the powerplant.
5 Comparatively little finished stock needed for service and repairs.
6 Ability to turn capital more often for a given output of complete units.
7 Minimum of accounting and inspection costs due to small number and simplicity of parts.
8 Minimum cost of individual tests of component parts and the final test, inspection and adjustment of the completed car.
9 Great reduction in labor, which, with the present shortage, is of paramount importance.



Cross-Sectional View of Doble Boiler
Steam side showing economizer sections filled with water and generating sections filled to normal water level

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THE CASE FOR THE STEAM CAR

By John Sturgess*

We believe we can best serve the members of the Society by answering three questions: why we build steam cars, why we build our particular type of car, and by what process we have eliminated other types. While *our* aims and activities have been completely lost sight of by gas-car builders, we have naturally been fully cognizant of *their* aims and activities, as was unavoidable from their prominent position.

Why We Build Steam Cars

The first answer, then, to the question, why we build steam cars, is, because we have seen in the public demands, and in efforts to meet those demands, an unmistakable longing for a performance which we, knowing internal-explosive cars as well as steam cars, believe steam *alone* can give—an ideal performance, to reach which steam is not only the shortest, but, indeed, the only route.

To conceive the performance which public and manufacturers alike are longing for, it is necessary we believe to abandon the standard of smoothness, flexibility, range of power, responsiveness, sense of security and command, which it is now habitual to accept, and to picture a standard set upon a far higher plane. The public wants a car with a soft, smooth, gliding motion, without noise or vibration at any speed; without clutch or gear-shift; without overheating in the summer or overcooling in the winter; with the smallest possible number of cylinders, moving parts and lubrication difficulties; with no possibility of stalling; with stored power built up in advance and replenished as fast as used, ready for instant application to the driving wheels; with ratio of car speed to engine speed always constant; with facilities to apply the maximum driving effort under the most adverse conditions; with one-finger control; and with kerosene fuel.

We believe that the true measure of any car performance can be classed under the following headings:

Range of power,
Smoothness of propulsion,
Ease of control.

*Stanley Motor Carriage Company

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Its desirability is further to be measured by its
Durability,
Reliability, and
Reasonable economy of operation.

Our contention that cars having steam powerplants are inherently superior to those having internal-explosive powerplants rests on the following statements:

Range of Power

The general terms flexibility, reserve power, hill-climbing power, acceleration and speed, cover the manifestations of the first quality, range of power. We use this term in a wider sense than range of horsepower, for the question of reverse power is involved. Horsepower is measured by three factors. It is force exerted over a given distance in a given time. Of these three factors, force only is what the motorist has in mind when he speaks of reserve power—the driving force. This is the only factor which he can control. When he applies this by opening the throttle, speed, and consequently horsepower, may or may not result.

Range of power, therefore, becomes range in the magnitude of force applicable to the driving-wheels. Reserve power becomes the difference between the driving force ordinarily utilized and that which can be utilized at any moment. Both are the direct result of range of effective cylinder pressures, with given gear ratio and piston displacement. One of the prime essentials of comfortable motoring is the sense of security from the consciousness of reserve power that may be instantly applied; hence he will compare steam and gas cars in this respect.

Reserve Power

The maximum mean effective cylinder pressure in the gas car averages about 100 pounds. The steam car has in reserve cylinder pressure well up to 500 pounds. Applying gear ratios and piston displacements to both types to obtain net driving effort or force, we will compare a 3⅛ by 5⅛-in. eight-cylinder and a 3 by 5-in. twelve-cylinder gas car with a 4 by 5-in. two-cylinder steam car. Gear ratios will be about 4½ to 1 in the gas cars and 1½ to 1 in the steam car.

The eight-cylinder engine will have a piston displacement (omitting non-working strokes) of 77 cu. in. per foot of car travel and the twelve-cylinder will have one of 104 cu. in. The steam car will have one of 41.4 cu. in.

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Multiplying these values by the maximum effective cylinder pressure, we obtain the following indices of driving force:

Gas car, eight-cylinder (3⅛ x 5⅛)..... 7,700
Gas car, twelve-cylinder (3 x 5).....10,400
Steam car, two-cylinder (4 x 5).....20,700

From this it will be seen that the range of driving force, and consequently the reserve power in the steam car are greatly in excess of either of the two cars.

The above calculations with gas cars are based on direct drive (high gear) because the accepted standard of their performance is what they will do on high gear.

Of course, the gas-car owner can shift to the low gear for greater power at reduced speed. But every time he does this he concedes the claim that the steam car possesses superiority in not having to do it; moreover, he himself will do it only as an emergency measure.

The gearshift has well been called a makeshift. It is a most unsatisfactory solution of the problem. Every motorist is demanding a better one, and every manufacturer is trying to find it. The fact that no gearshift is needed with a steam car is one of the strongest reasons why we build such cars.

It is not claimed that the steam car can maintain the above high tractive effort indefinitely at high speed. Such would not be necessary under any road condition ever likely to be encountered. Its temporary availability is all that is desirable. Even with 400 lb. mean effective pressure at 30 m.p.h., such tractive effort would amount to 80 hp.; at 50 m.p.h., it would equal 130 hp. This great amount of power is available for as long a period as road conditions would usually permit its use.

It is probably true that many stock gas cars, including the typical cars cited, could maintain a higher rate of speed on an ideal track, say on Sheepshead Bay Speedway. But what proportion of the power the gas car develops on the track is available on the road? What is the value to the motorist of a track performance which cannot be duplicated on the road he is going to use?

It is not that steam cars cannot be built to meet track conditions, but in order to maintain a high average road speed, with acceleration inconceivable to the gas-car driver, the steam car does not have to be equipped with a powerplant capable of producing an amount of power that can rarely if ever be used continuously.

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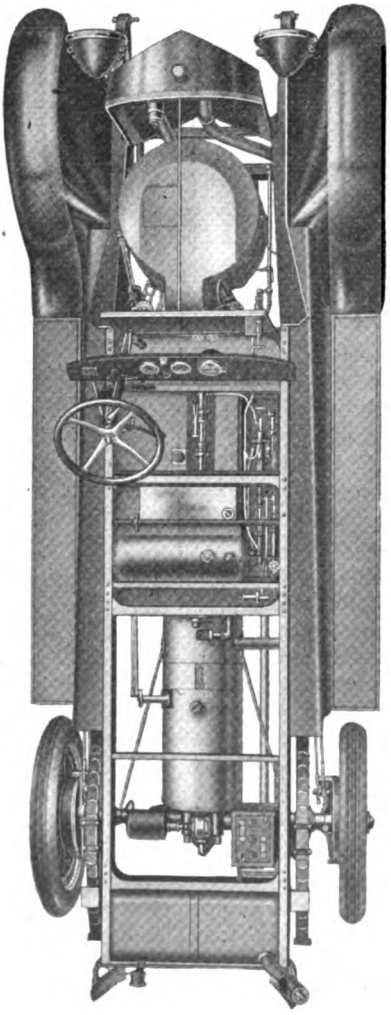
There is a certain subtlety to this that can best be realized when a car known to be capable of say 75 m.p.h., in ordinary road going, at 40 m.p.h., is suddenly



CHASSIS OF THE STANLEY STEAM AUTOMOBILE SHOWING POSITIONS FOR POWERPLANT UNITS

passed by a steam car which has been trailing for miles, the steam car suddenly shooting ahead, within the space of a few rods, at the rate of 60 m.p.h. or more. From our experience we know that no car can show such accelera-

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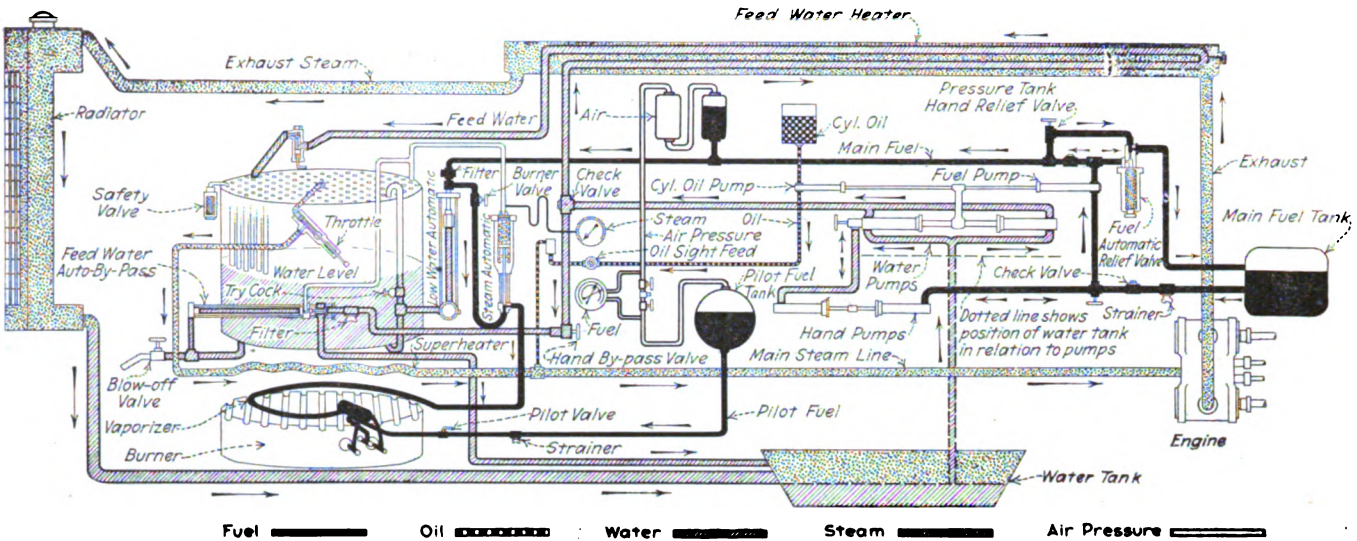

DIAGRAMMATIC PIPING LAYOUT OF THE STANLEY STEAM AUTOMOBILE

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tion unless it is capable of 90 m.p.h. at least. This is what misleads many people to think that the Stanley is the fastest car in the world. It is not, by any means, if we mean indefinitely maintained speed. It does not have to be to meet the conditions of motoring that the driver is faced with every day and wants most to overcome.

Smoothness of Propulsion

This range of power, with its method of application, by gearing the engine directly to the rear axle, which in practice would be impossible in the gas car, results in a smoothness of propulsion which is particularly apparent at moderate speeds, especially during heavy going. It arises from the fact that two double-acting steam cylinders produce a perfectly uniform continuous turning moment on the crankshaft.

At high car speeds this smoothness is also gratifying. The engine speed is always in identical ratio with the car speed, and at 60 m.p.h. the engine is running at but 864 r.p.m., with 3456 piston reversals a minute, compared with say 2500 r.p.m. and 40,000 piston reversals a minute in an eight-cylinder gas car.

Ease of Control

In steam cars, control is far more completely centered in the single-throttle lever than gas-car control is centered in the gasoline throttle. It is even far more completely centered than in the gas throttle *plus* the gearshift; and in the steam car the gearshift is completely eliminated, and no substitute is needed. The single throttle-lever suffices to start the car from rest and to control the power from zero to 500 per cent above rating. A few inches movement of one lever covers the entire range. A still further increase in power is obtained by releasing a pedal, but this is only needed on the rarest occasions. Reverse is effected merely by pressing the above-mentioned pedal forward, if desired, while the car is still running forward.

Turning to the mental aspect or degree of driver's attention, the only thing the steam driver is cognizant of is the speed of the car. Car speed and steering are all he has to think of. The mechanism of the car can be completely dismissed from his mind.

The gas-car driver, on the other hand, has to give a certain degree of attention to engine speed, to the extent of almost complete absorption when starting, or in heavy

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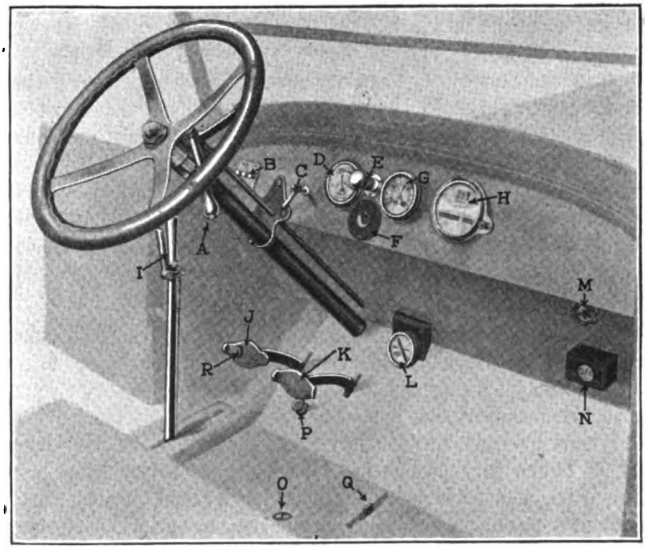
traffic. One condition governs car speed, other conditions govern engine speed. The driver must maintain a mental picture of both so as to be alert to adjust their relationship, and this adjustment involves action by his two feet and one hand or one foot and two hands, and he must do it *at once* or he is too late.

Ease of Starting

It is commonly supposed that in starting the steam car is at a great disadvantage, requiring much preparation and considerable time. This is a very erroneous impression. A true comparison of the preparedness for instant motion of steam and gas cars is to the advantage



INSTRUMENT BOARD AND CONTROL DEVICES

A—Throttle
C—Main burner control
D—Steam gage
G—Fuel pressure gage
J—Reverse pedal
K—Foot brake
L—Water level indicator
M—Hand bypass valve
N—Sight electric indicator
O—Water tank gage
P—Horn button
Q—Hand pump lever stub

of the steam car. For such comparison will embrace all starts occurring all day and every day, not simply the individual case when the steam car is cold, this constituting not 5 per cent of all starts made. It will then be apparent that the normal condition of the steam car is one of readiness for instant start, even before the driver has taken his seat, a single movement of the throt-

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tle lever being all that is needed. This is true for probably 95 per cent of all starts made, arising from the simple fact that steam is up *all* such time, and is to be contrasted with the well-known sequence of operations in starting a gas car, involving time, close attention, dexterity, and not infrequently unpleasant noise.

Even in the initial daily start, after the car has been standing for a considerable time, the facility of starting is hardly less, for the pilot has maintained a good head of steam, and it is necessary to open up the main burner only a moment before starting.

It is only when steaming up from cold, which need not occur once in several weeks if the pilot is kept burning (as it usually is) that the steam car may be behind the gas car in starting. But if, as in most cases, the time at which it is desired to start can be anticipated by 5 to 10 minutes, and the burner started accordingly, there need be no delay. It is true it used to be a somewhat irksome matter to set the burner in operation, just as it used to be even more irksome to crank the gas car by hand; but today it is only necessary to press a button to heat and ignite the pilot, and then open the main burner valve. Steam is available a few minutes later.

Cold-Weather Starting

The foregoing comparison of starting facility would not be complete if reference were not made to winter conditions. The zero morning in the cold garage causes no apprehension to the steam motorist. There is no fear that he may have a ten-minute fight with his engine, nor is there any consequent leaking of unwelcome gasoline into a crankcase. The long, frigid waits at the curb are perhaps more dreaded by the gas-car motorist than anything else. Only short runs are made in cold weather, with long stops. Stored heat is the best weapon to fight this difficulty. That is why the gas motorist is more watchful to have his radiator full in the winter than in the summer. In the steam car there is a greater volume of stored heat, and it is better distributed than in any other car. The steam motorist does not have to enrobe his radiator at every stop, for it presents no thin sheet of water to the air. It is empty the minute the car stops.

The steam car does not freeze for the same reason that the building provided with a hot-water heating system does not freeze; but whereas the building has chilling windows, all susceptible parts of the car are heavily

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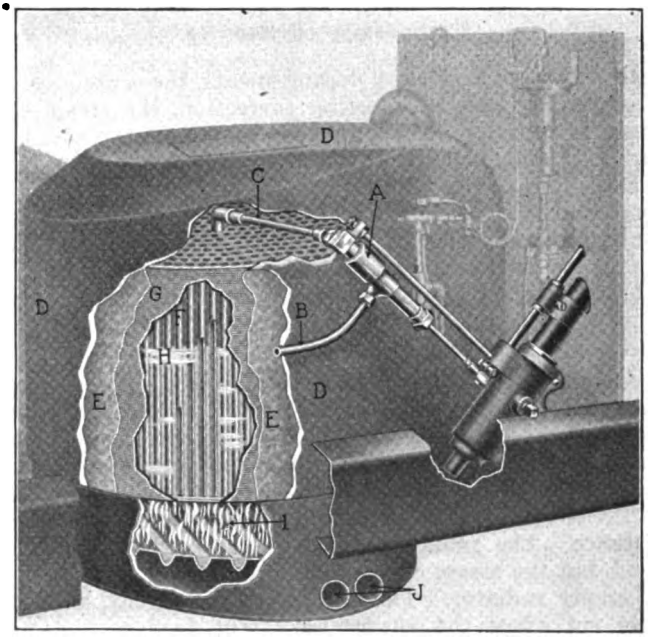
clothed with insulation. Consequently the small pilot, always burning, maintains the heat. If there is objection to long-continued burning of the pilot, a small city gas jet with hose connection to the burner suffices, and costs little. Finally, the water in the boiler can be drained with little effort, or even be permitted to freeze solid with but slight danger, and that to small parts only.



SECTIONAL VIEW OF BOILER

A—Throttle
B—Outlet to superheater
C—Steam outlet from boiler
D—Asbestos covering
E—Galvanized iron covering
F—Boiler tubes
G—Piano wire
H—Water level
I—Main burner flame
J—Main burner mixing tubes

Durability of the Steam Car

Deterioration by age and use of the steam powerplant does not appreciably affect car performance. In other words, it is more enduring. This is due to the low-speed engine, completely encased, of few moving parts; to absence of dust taken into the cylinders (characteristic of all internal-explosive engines); to simplicity of the direct connection to driving wheels, without the complicated transmitting devices of the gas-car clutch, jointed shafts, and gear boxes; and, in fine, the less sensitive steam-power cycle.

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Durability of the boiler is not dependent on wear in the ordinary sense, for it has no moving parts. At the same time, deterioration naturally occurs. The normal life of a boiler is from four to five years. Replacement does not, therefore, result in an appreciable annual repair charge, especially in view of the low cost of other repairs.

Examples of Reliability

In respect to accidental derangements, the consequences of such, and ease in effecting correction, the steam car again is superior. Local failures may lame the car, but rarely, if ever, halt it. It can always get home, even if it has to limp a bit. Steam cars have been driven home comfortably on one cylinder, the other having a broken connecting-rod, without stopping to make repairs. Engine parts have worn half an inch (through long-neglected lubrication) and the driver hardly knew it. Boilers may start to leak, but the car can still be run hundreds of miles before the leak is likely to become serious.

Automatic features may conceivably fail, but the driver can still operate under hand control. The fuel tank may become unexpectedly empty, but there is no dying gasp without notice. The steam car can run several miles without fuel by drawing on the stored power, and kerosene (the fuel) can usually be obtained within that distance. The radiator may leak faster than it can be filled, but the steam car cares nothing for that; it enjoys an empty radiator. The battery may give out, but that does not affect the powerplant. The feed pumps may conceivably fail, but the supplementary hand pump may be used in such emergency.

Advantages

It will now be well to state why the essential features of our burner, boiler and engine have been adopted.

Burner—We use the vaporizer system, the fuel being vaporized by heat and burned on the bunsen-burner principle. We do so because we consider it outclasses other systems in respect to weight, space, stability, security, automatic regulation, the conservation of stored power, simplicity of apparatus and starting facility.

Minimum weight results from the burner comprising only a vaporizer tube with nozzles, mixing tubes and burner chamber, the total weight of these parts being 65 pounds.

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Minimum space results from the short flame length and intensity of combustion.

Stability results from the fixed conditions, owing to absence of all moving parts.

Security results from the closed combustion chamber and small amount of fuel contained within the burner at any one time.

Automatic regulation is facilitated by the ease of controlling the rate of combustion.

With regard to the last claim, starting facility, we refer to ease of starting the burner, not the car, for, as already explained, the burner does not have to be started once in a hundred times that the car is started, owing to the steam pressure continuously maintained by the pilot. But on those few occasions the burner is started by simply pressing an electric button.

Boiler—We have adopted the fire-tube drum type of boiler, because we believe it satisfies the requirements of stored power, ample steaming capacity, large disengaging surface, dry steam qualities, positive control, small size and weight, endurance, constructional facility and absolute safety.

Its great stored power results from its large water content, maintained at full temperature. It contains 60 lb., which at its temperature of nearly 500 deg. fahr. represents 1,500,000 ft. lb. of available energy.

Its ample steaming capacity results from its 104 sq. ft. of heating surface, afforded by 750 half-inch tubes.

Its large disengaging surface results from its large diameter and shallow depth—2 sq. ft. to disengage the steam generated in 8-in. depth.

Its dry steam qualities result from this disengaging surface, the liberal steam space, the high pressure, and the drying effect of the fire tubes within the steam space.

Positive control results from the definite water level and large water content.

Its small size results from the absence of idle space. It is but 23 in. diameter and 14 in. deep.

Endurance results from its simplicity of design, substantial proportions, and low intensity of action.

Constructional facility results from standardized and familiar manufacturing processes employed.

Absolute safety results from the fact that the boiler simply cannot explode or even rupture, with any dangerous result. Long before the ultimate strength of the wire-wound shell is approached the small tubes col-

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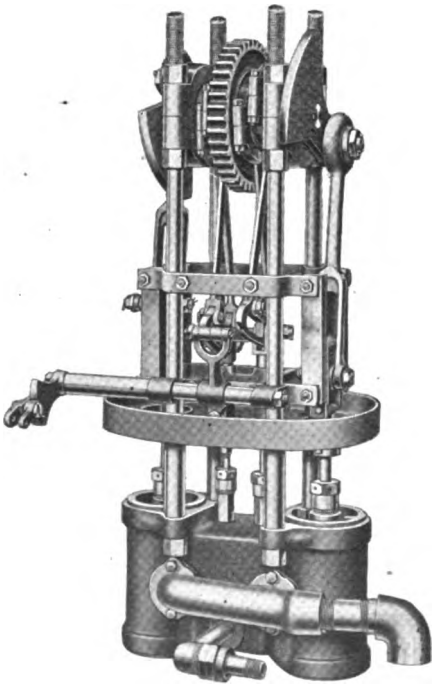
lapse in such a manner that the escape of steam is shielded, and rendered sufficiently gradual to obviate all danger.

Engine—We use a two cylinder, double-acting, slide-valve engine, 4 by 5 in., with Stephenson link valve motion, straight crankshaft carried in large roller bearings, overhung single-web cranks at 90 deg., and roller bearings on crankpins. The engine is permanently geared to the rear axle, without clutch or transmission, the gears being plain spurs of large diameter set in the center of the crankshaft, between the bearings. The gear ratio is such that there are one and one-half turns of the engine to one turn of the driving-wheels.

The prevailing pressure is between 100 and 200 lb. per sq. in., but may easily be run up to double this, or even to full boiler pressure for rare emergencies. Normal cut-off is at ⅜ stroke, but may be changed at will to ⅝ stroke. The range of tractive effort resulting from this, notwithstanding the low gear ratio and two cylin-



TWO-CYLINDER STANLEY STEAM ENGINE

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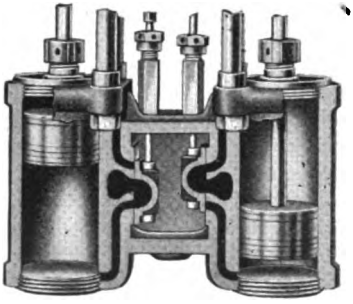
ders, is, as already stated, double that of the average high-power gas car.

The engine weighs 265 pounds. A stock engine under test-block run has frequently operated with a steam consumption of 16 to 17 lb. per hp. hr., a remarkable result from so small an engine.

Such an engine combines all conceivable requisites of smoothness, great power range, uniform turning effort, few moving parts, light weight, long life, economic operation and rugged simplicity.



SECTIONAL VIEW OF CYLINDERS AND STEAM CHEST

ELIMINATION OF OTHER DESIGNS

Other designs of engines have been eliminated by keeping in view the basic requirements enumerated, and checking deductions by repeated tests on alternative designs.

With burners, the system using air, steam, and mechanical means for atomizing the fuel has been extensively experimented upon, but according to our experience does not satisfy all the requirements of automobile needs as well as has the vaporizer system. An intense combustion is obtainable, but in respect to weight, size, simplicity and ease of control, stability and maintenance of stored power when the car is standing as well as in numerous practical difficulties, the atomization system has not, thus far, matched up to the vaporizer system we employ. The atomizer requires either a steam jet or considerable additional apparatus, such as power-driven blowers. The burner must be repeatedly extinguished and re-lighted in the process of regulating steam pressure, and maintain this pressure when the car is standing; this involves starting up the power-driven blower and electrically igniting the gases, obviously an undesirable condition while the car is unat-

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tended. This is not to be compared, in point of safety, with the maintenance of a small, continuously burning pilot, and a graded combustion of the main burner.

The flash-type boiler has not been thought worthy of serious consideration, its sole claims being that it will steam up quickly and that its thermal efficiency is good. It completely lacks the prime essential of stored power, and its control is complicated and highly uncertain, in that both temperature and pressure have to be regulated.

The water-tube type may be regarded as a compromise between the flash and drum types. Its principal claim is that steam can be raised quickly, owing to small water content, but this obviously sacrifices the paramount advantages of stored power and overload ability, which are directly proportional to water content. The pilot and vaporizer system of combustion maintains steam pressure at practically all times, and the occasions when quick steam raising would be beneficial are so few as to render this nominal benefit of no practical advantage. In 99 out of 100 cases the car with a drum-type boiler and pilot would be far down the road before the car with the water-tube boiler without pilot would have got up steam, although in the one case in a hundred the latter might be a few moments ahead.

The water-tube type is more difficult to control as regards water-level; in fact thus far it has not been satisfactorily controlled in this respect at all. Furthermore, the control problem involves more complication, which is undesirable. Water-level and circulation are quite uncertain owing to weakness of the parallel circulation and its tendency to reverse, resulting in the so-called "geyser effect" and danger of burning the tubes.

A refractory lining is necessary in the casing of the water-tube boiler, which, while a minor objection, still constitutes an additional problem, especially when dimensions and weight are reduced to the extent they must be to suit automobile needs, where weight and space are at a premium.

As the engine is a relatively minor problem in steam cars—owing to steam engine practice being so highly standardized—difference of type or details of construction are not of such pronounced moment as the major problems of burner and boiler. Another company (now among the gas-car manufacturers) for a few years used a compound engine. The unaflow type has been proposed and much discussed recently. We conducted tests on a

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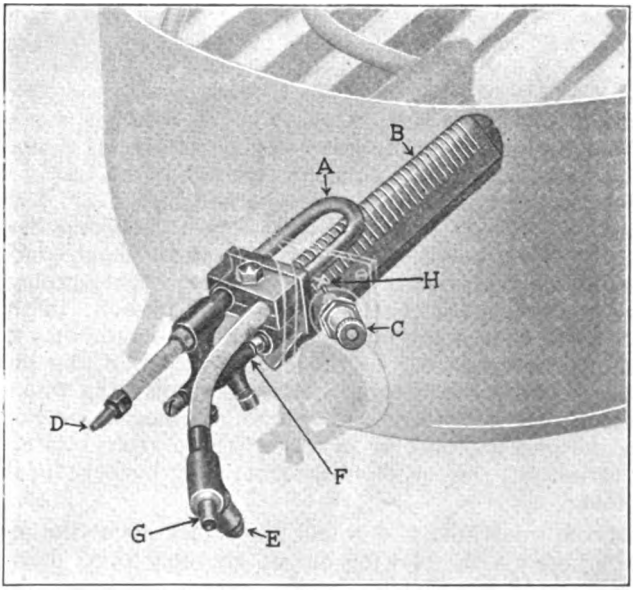
unaflow engine a year ago and failed to discover any superior steam economy. We did encounter many minor disadvantages in this type when applied to automobiles, among which were uncertainty at times as to whether the car would run forward or backward on starting, and the possibility that it would not start at all. Furthermore, its high compression was a disadvantage and its weight and size exceeded that of the contraflow type.



PILOT CASTING ASSEMBLY

A—Vaporizing tube
B—Pilot casting
C—Spark-plug
D—Return connection to switch
E—Fuel inlet
F—Pilot nozzle
G—Connection from battery
H—Spark gap

These difficulties could be overcome in considerable measure, but so doing would modify the unaflow type to the point that its characteristic feature would be largely lost, and there would be added complications in construction.

The use of high superheat, moreover, largely minimizes the benefit of the unaflow cycle, and the absence of vacuum further defeats its primary advantage.

AUTHOR'S CONCLUSION

The foregoing states briefly the case for the steam car, as we view it. We have said nothing of the gasoline

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situation, which it is well known is causing concern in the automotive industry, both as to quality and quantity, a concern that promises to become deeper with the increasing use of automobiles and particularly the most extensive increase of trucks, tractors and airplanes.

The time has come, we believe, when the industry must pause to consider seriously our assertion that steam is not only the shortest but indeed the only route to automobile perfection.

THE DISCUSSION

HUGO C. GIBSON (M. S. A. E.): If steam is a good propulsive means, let the steam exponents give us a steam plant for propelling airplanes through the air. It would have fewer parts, as the automobile steam plant has. The problem is really only one of utilizing heat, and I believe that the efficiency of a steam plant can be even higher than that of an internal-combustion engine, if loss of heat is prevented by proper insulation; in this particular respect the internal-combustion engine has the disadvantage that a very large proportion of the heat must be thrown away by arranged means—the cooling system. In the steam plant arranged means for holding the heat in must be provided; there is definite conservation of the heat units from the burner to the cylinder.

MR. SCHLESINGER:—I would like to ask about the provision against the blowing out of the pilot light; does it happen? If so, what is the result?

I understand that the electric form of vaporization is new this year in the Stanley car. What is the current consumption of this heating coil?

I would also like to ask for the definition of a unaflow engine. Reference was made in Mr. Doble's paper to priming. We all know what priming means in gasoline practice, especially after this cold winter, but I have never before heard it applied to steam engine practice.

Steam versus Gas Engines for Airplanes

MR. EMORY:—I would like to answer Mr. Gibson's question about the use of steam for driving airplanes. The conditions required in an automobile are maximum torque with minimum speed; whereas the conditions in an airplane are approximately constant torque and constant speed. Of course, the latter is a condition for

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which a steam engine is not particularly well fitted—at least, not better fitted than a gasoline engine.

In regard to the question asked about the pilot light, I do not believe that a pilot light will be blown out by a blast of air, but unfortunately it is frequently blown out because of absence of air.

I have driven the steam car a good deal—at present I am driving a gas car—but I must say that there is no comparison between the two. I would rather have an old Stanley car than a brand-new twelve. If the Stanley Company will only make a boiler that will not give out at unexpected moments, and a pilot light that does not go out just as it is needed, I think they will have what they refer to as the "perfect automobile."

Scale Formation in Steam Boilers

BENJAMIN F. TILLSON (M. S. A. E.):—The questions I have apply more particularly to powerplant engineering, as a steam engineer sees them. A comment was made that 90 per cent efficiency was obtained in one of these steam plants. That is a result so unusual for steam plant engineering as to appear impossible because of the loss of 15 or 20 per cent of the heat units in the fuel, depending upon the temperature and volume of the gases leaving the furnace. I assume that the 90 per cent rating was in terms of the heat units in the actual boiler horsepower produced as against the heat units obtainable from the fuel used? The question of the possibility of scale forming in the boiler appears to me to be of more serious moment than this meeting has been led to believe. I would like to learn why the practice, generally found necessary in powerplant engineering, of eliminating all possible oil from feed water, does not also obtain in the practice of steam engineering as applied to small automotive units. If we do not eliminate oil from our boilers we get carbonized oil, which forms in the boiler tubes. Overheating and distortion of the boiler tubes result from the retardation of heat transmission caused by this deposit and disastrous failures have been due to this cause.

What provisions are there in steam cars for cleaning scale which does form in boiler tubes?

Answering the question raised in regard to the term "priming" in steam boiler practice, it is considered to be the mechanical carrying off of water with the steam and is extremely objectionable.

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Four Companies Developing Steam Cars

MR. BROWN:—There are now, to my knowledge, four different companies making progress in the development of the steam car. I say development, because all of them are capable of great improvement. Of those companies, we have heard of two; a third is the Star Motor Company of Chicago, which has in operation on the road at the present time a steam truck that differs from the two described in that it has a water-tube boiler somewhat similar in general type to the Babcock & Wilcox. The combustion mechanism is similar to that of the Stanley. The other car has been designed and built at the Doble Laboratory of San Francisco by brothers of Abner Doble of Detroit, who originated the lighting of the fire by spark-plug. The boiler of the San Francisco Doble car is somewhat different from any of the other three. It is practically a very small circulating boiler with a very large economizer. The San Francisco Doble burner does not require any air pressure, has no small orifices to clog up, and offers some advantages.

The reason stationary powerplants have trouble with oil and these automobile plants do not is owing to the difference in the character of oil used. If we used the same class of oil in automobiles that is used in powerplants, we would burn our boilers inside of two or three days. The use of high-grade engine oil in boilers is not as harmful as the use of animal and vegetable oils.

Unaflow Engine Development

The definition of a unaflow engine was requested. In 1850 an engine was invented in which steam was admitted at the end of the stroke and exhausted at the middle, but it was not called a unaflow engine. Other inventions along that line followed until 1880; but none of them were called unaflow engines. In 1908 Johann Stumpf developed an engine of similar type, in which he achieved great economy such as had not been accomplished by any of the others; he called it a unaflow engine. That is the unaflow engine which has made a reputation for itself. With that definition of this type of engine, none of the engines described, as constructed by Doble or Stanley, are unaflow engines; they are simply copies of the 1850 engine which never was a success and which never was put on the market.

I am sorry to hear such claims of perfection, because I do not believe it does the cause any good to claim greater perfection than can be achieved in practice. Every one

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knows the advantages of the steam engine and every one who has had anything to do with it also knows the shortcomings of the steam combustion apparatus and boiler for automotive practice; great strides are being made to overcome these difficulties and when they are overcome—which I believe will be in the near future —then steam power is going to take its place.

S. N. CASTLE (M. S. A. E.):—Most of the claims in the papers have been commercial or sales rather than engineering arguments. Two points however do not appear to have been contrasted. Doble claims to get rid of the boiler scale by means of an oil film. What happens to the scale on the Stanley car? Second: Ability to run 2500 miles on one charge of water is claimed for the Doble car, while the Stanley people state that their car can run from 150 to 200 miles at the outside, on one charge. From what has been shown in the papers, the conditions of condensing are practically alike in the two cars with water charge apparently in favor of the Doble car.

EARLY STANLEY STEAM CARS

PRESCOTT WARREN (A. S. A. E.):—When we started in business in 1896, Mr. Stanley had the same great idea as Henry Ford, that an infinite number of people would pay five or six hundred dollars for an automobile; so we put our first cars for sale in 1898 at a price of $600. At that time Haynes and Duryea cars were selling for from $1,200 to $3,000. In 1910 we built a 10-hp. car, with a speed of 60 m.p.h., and sold it at $850. That car was not equipped with a windshield, to be sure, and many of the accessories that people now demand. It shows, however, what can be done in the matter of price. We were building three different sizes of cars. In 1914 we decided that it would be better business to concentrate on one model. Our demand showed that the 20-hp. car was the most popular. There is no doubt in our minds that the steam car can be built of the same quality and in the same production for less money than any other car.

MR. STURGESS:—Mr. Schlesinger inquired about the current consumption required for starting the pilot light. About 200 amp. for 15 to 20 sec. is the maximum flow. This is about the same current flow used on the average gasoline car, but the battery is used only very rarely—certainly not more than once a day—so that the net load on the battery is very much lower.

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Life of the Stanley Boiler

Mr. Emory asked why we do not make a good boiler, and referred to its life. This compares well with the life of many vital parts of gas cars. It should be borne in mind that the boiler is about the only part of the steam car that wears out—the engine is practically indestructible, and there is no transmission.

Importance of Using Proper Water

In any engineering field there are occasions where we have exceptional cases of trouble, but it is erroneous to regard those as typical. There was one such case where a man had a boiler he could not make last for more than three months. We had to put in a new boiler, which again failed. The case looked so bad that we sent a special engineer to investigate. We knew that there were many other Stanley users in that same town, and they had no trouble. We finally found that he was using water from a well that was full of magnesia and lime, and he never blew off his boiler, so of course he had trouble. He ceased to use that water and had no more trouble.

Kerosene Mixed With the Lubricant

As regards oil causing trouble in the boiler, we as engineers are all thoroughly familiar with the old axiom that at all costs oil must be kept out of steam boilers. It is well known, however, that the practice of putting kerosene in boilers acts as a deterrent to the formation of scale arising from the inclusion of oil and while it is not very practicable to put much kerosene into a large boiler, yet I believe the practice is not adopted as much as it might be. We make a practice of mixing kerosene with our lubricant and as a consequence the oil—irrespective of what it ought to do, does not cause any trouble. I attribute this to the fact that the kerosene gets on the tubes first and the oil does not have a chance to adhere. Of course, we make a practice of blowing off fairly frequently, and I think those two facts account for the circumstance that in practice we do not have any trouble from oil in the boiler.

Efficiency of the Stanley Powerplant

In tests on our 4-by-5 stock engine, working with about three-eighths cut-off, and using superheated steam, we have frequently obtained steam consumptions of

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from 16 to 17 lb. per hp. With a condensing engine running under 28 in. of vacuum, a rate of 13 to 14 lb. of steam per hp. is considered very good. I attribute this performance to the high superheat, to the insulation preventing radiation, and to one or two other minor constructional causes. This performance works out to 77 per cent efficiency, based on the Rankine cycle. Those tests were carried out under strict scrutiny with calibrated instruments, and independently checked by many engineers.

Water Tank Capacity

The figures of water consumption (miles per tank of water) which we gave, we regard as conservative, and those which will be experienced every day by the average automobilist. We do not know whether Mr. Doble's figures are on the same basis. The Doble design uses a fan to augment the condensation. We carefully considered this, but it was regarded as an added complication, involving more parts to wear and greater cost. From 150 to 250 miles on a tank of water is about all that is really required, because people do not often go very much farther than that in one day; and when they are in the garage there is no objection to refilling the water tank.

R. H. THAYER (M. S. A. E.):—The meaning of the term "unaflow" can be explained by comparing it with "counterflow." In the unaflow type of engine the steam enters the end of the cylinder through a port in the usual manner and forces the piston ahead, but instead of exhausting this spent steam by causing the piston on its return stroke to drive the steam out through the same port through which it entered (thus compelling it to counterflow) the live steam in the unaflow cylinder follows up the piston and is exhausted through ports in the cylinder walls located at about its center of length; these ports are uncovered by the receding piston head.

As to the real merits of the counterflow and unaflow types of engine, much can be said in favor of each, especially with respect to their adaptation to steam-propelled vehicles. The determining factor with us, thus far, has been one of expediency as prompted by physical conditions. Since we are using with highly satisfactory results, the unaflow type in passenger cars and the counterflow type in our steam trucks, we are committed to neither.

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Problems in Doble Combustion Mechanism

The idea of means whereby the pilot light might for once and all be eliminated involves something more than the mere means "of lighting the fire by spark-plug," the successful and repeated ignition, by spark, of sprayed kerosene mixed with the enormous volume of air necessary to insure combustion to the exclusion of soot, smoke and smell. Nor did the problem cease with means for initiating combustion in so rarefied a mixture. It was found that with the burning of a sufficient quantity of fuel mixed with the proper amount of oxygen to produce the necessary temperature whereby the excessive and sudden demands for steam would be satisfied, an enormous volume of flame resulted. It was the confining of this flame in the necessarily limited space in the fire-box, 20 in. square and 8 in. deep, that proved the greatest problem. As we review the efforts of engineers, it becomes evident that it was this task of properly confining the open flame without interfering with proper combustion that led to vaporizing the liquid, producing a gas which was at once readily ignited and easily confined in the necessarily small fire-box such as could be adapted to a steam-propelled vehicle. This pre-gasification, however, necessitated the use of means for preheating, including the troublesome pilot light.

Lubricating with High Steam Temperature

In regard to the action of lubricating oil with the steam, Mr. Tillson lost sight of the fact that we mentioned particularly that gas-engine cylinder oil was used. Owing to the high temperature and pressure in our boiler, this oil does not exist as a liquid, but rather is present as a gas. In going with the superheated steam to the valve chest and cylinders the temperatures become low enough to condense a part of the oil to a liquid, which properly lubricates the moving parts. Passing with the exhaust steam to the condenser, both steam and oil are converted into liquid form and are fed again into the boiler.

Over-all Boiler Efficiency

Mr. Tillson's question as to the basis on which we

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arrive at a figure of 90 per cent over-all boiler efficiency can be answered as follows:

Water evaporated, lb. per hr	630
Superheat, deg. fahr	25
Total heat (Mollier diagram), B.t.u. per lb	1245
Temperature of feed water, deg. fahr	62
Total heat above feed water temperature, B.t.u. per lb. = 1245 — 62 =	1183
Fuel burned, lb. per hr	41.24
Heat value of the fuel, B.t.u. per lb	20,100

$$\text{Efficiency} = \frac{1183 \times 630}{41.25 \times 20100} = \frac{74600}{82700} = 90 \text{ per cent}$$

STEAM CARS

By Prescott Warren*

The conception of a self-propelled vehicle, of what we to-day call an automobile, is one of the oldest in human annals. It is probable that that revolutionary being who, when mankind was emerging from savagery into barbarism, first applied the wheel to a vehicle, had vague glimpses in his mind of an occult power which would propel that vehicle without the aid of man or animal.

All through the myths and legends of the ancients we find conceptions of the self-propelled vehicle, that swift, silent, mysterious carriage which would transport them with speed, comfort and safety on the land or in the air. But so remote did this seem, so far beyond the grasp of mankind, that the ancients had not the audacity to claim it for themselves even in their legends and myths, but attributed it to the gods.

As the Egyptians, the Persians, the Greeks and the Romans attained their high degrees of civilization the desire for this godlike device became more and more persistent. Civilization realized instinctively that somehow mechanical forces could be set to work to annihilate time and space, but throughout ancient times the wish remained only a wish.

*President, Stanley Motor Carriage Company

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Leonardo da Vinci's Self-Propelled Vehicle

The first record of a concrete human conception of a self-propelled vehicle is found in the fifteenth century, when Leonardo da Vinci, one of the greatest, and certainly the most versatile of geniuses that ever lived—poet, philosopher, architect, sculptor, painter, engineer, statesman, warrior—conceived a military vehicle propelled by steam for assaulting the walls of fortified towns. It was, in effect, what we would now call a tank. Leonardo not only conceived it, but designed it, and although he never built it he left a full set of drawings of it. Modern engineers have examined his drawings and have found that the vehicle is entirely practicable, that it could be built and could be made to run.

The first real impetus toward the attainment of civilization's most crying need came with the invention of the steam engine in the last quarter of the eighteenth century, less than 150 years ago. More progress has been made in swift transportation in that period than in the 60 centuries of recorded history before it. Immediately upon the discovery of this new force earnest attempts accompanied by heart-breaking failures were made to adapt it to propelling land vehicles. The attempts became so numerous, in fact, as to attract the bitter opposition of the Tory or reactionary element in every country, that element which always wishes to let well enough alone. The opposition was so effective that it resulted in legislation ruling such vehicles off the road. It has been said, and probably truly, that this legislation set back the development of the automobile, and thereby the advance of civilization, by a full century.

Let us pause here to pay tribute to the indomitable perseverance of those hardy pioneers—business men and engineers—who, when they found themselves ruled off the highways, had the audacity to conceive of buying their own rights of way, building their own graded roads and laying rails upon them, and thus set the beginning of our modern railroad system; who not only had the imagination to conceive this stupendous thing, but the courage to put their capital into it and the brains to execute it.

Perhaps we should pay even higher tribute to those other pioneers, those other industrial radicals, who foresaw that the self-propelled vehicle running on rails on a graded road, on a private right of way, would not, by any means, even begin to meet civilization's crying de-

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mand for a vehicle that any individual could drive himself wherever he wished on a public highway. Luckily, nature endows certain men in every epoch with that pioneer instinct which impels them in the face of laws, ridicule and mechanical discouragement, to achieve the seemingly impossible.

Automobile Development Retarded by Legislation

I have said that it is probably true that legislation set back the development of the automobile by a full century. Some exception may be justly taken to that statement. Remember, the first automobiles burned hard fuel —first wood and then coal—and they ran on steel tires. But the automobile as we conceive it today—a vehicle to transport us on the highways with speed, comfort and safety—awaited two things—liquid fuel and the pneumatic tire. Neither was available for many years. It may be permissible to discuss them in the order not of their advent but of their importance.

The automobile would be well-nigh impossible without the pneumatic tire. We cannot conceive of its widespread use without this which seems at first glance a mere accessory. It is only twenty-five years since the pneumatic tire was invented. But in that short time the automobile has revolutionized our social and economical life, and its manufacture has become America's third greatest industry.

Discovery of Liquid Fuel

The automobile was also of little value without liquid fuel. We, to-day, cannot conceive of putting up with wood or coal.

Liquid fuel was discovered about 1860. One of the first pioneers to experiment with it discovered that it had explosive as well as inflammable properties.

Another soon conceived the brilliant and almost revolutionary thought of inducting a gasified drop of liquid fuel into the cylinder of an engine, exploding it there, and allowing the resulting force to drive the piston down.

This was *power direct from fuel.*

The Fork in the Road

Here lay the fork in the road. The pioneers saw before them now two paths, each apparently beckoning them to their goal. Remember, up to this time all ex-

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periments on self-propelled vehicles had been with steam. What motives were these, then, which led the engineering fraternity down the wrong fork?

Unquestionably the lure of getting power direct from fuel was foremost.

"Why burn fuel under a boiler, transfer the heat to the water to turn it into steam, and then conduct the steam to a cylinder, when the fuel itself can be burned (exploded) directly in the cylinder, thus dispensing with the steam boiler and burner units altogether?" The thought seemed brilliant and feasible. It aroused the spirit of adventure—the instinct of the pioneer—in the engineers of the day.

Explosive Engine Characteristics

For certain purposes, indeed, it has great merit—for stationary engines and such uses as require only constant power and constant speed.

But engineers at once began to adapt this new engine, essentially a constant-speed, constant-power device, to the variable-speed, variable-power automobile—which is, perhaps, the most inappropriate use to which it could be put.

Patent Protection

Steam practice, even at that day, was highly standardized. Indeed, for self-propelled land vehicles driven by steam, Stephenson's types of boiler and engine are to day the standard. There was scant hope, therefore, that the engineer could develop anything in the steam line so revolutionary and so advanced that he could secure patent protection. But he could see in this new, unknown, untried device, so simple in theory and so alluring to his inventive instincts, a splendid opportunity to be the first to hit upon that secret in harnessing it to the automobile which might be so basic as to give him complete patent protection and yield him an enormous fortune. And as each engineer heard rumors of what other engineers in other parts of the world were doing this only spurred him on to swifter action, and encouraged him to think that if these others were struggling along the same path, it must surely be the right path.

It was immediately apparent that a clutch and a change-speed gear had to be interposed between the engine and the rear wheels. But such problems did not deter these courageous pioneers, nor did the more deli-

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cate problems of vaporization, carburetion and ignition. Persistent inventive genius, following the engineering fashion which had set in, combated the difficulties and almost overcame them. Nor did the engineers hesitate long to duplicate their internal-explosive engine, or quadruplicate it, or to put in six or eight or even twelve cylinders to drive one automobile.

Is not the theory of the internal-explosive engine as applied to a variable-power, variable-speed device like an automobile fundamentally a fallacy? The manufacturer of to-day will say he builds gas cars because the public wants them. The automobilist, who knows only what the manufacturer has taught him, will insist that gas cars must be better because there are more of them —that if steam cars were better the manufacturers would build steam cars. So the manufacturers blame the fashion upon the public, and the public blames it upon the manufacturers.

Internal-Explosion Engine Complications

Those early pioneers were really to blame—those who began with the beginnings of the explosive liquid fuel. They were carried away with the idea of getting "power direct from fuel," and lost sight entirely of the fact that their real problem was not merely to get the power from the fuel, but to get the power from the fuel to the driving-wheels of an automobile, which device, above all others, requires instantaneous variability of power, and instantaneous response to the driver's will. The complications introduced to make the internal-explosion engine deliver its power to the rear wheels became in time, indeed almost immediately, far greater than those of the steam boiler and burner units, which, with the best intentions in the world, the pioneers had undertaken to get around.

Clutch, gearshift, complete electric plant for ignition, and another complete electric plant for starting the internal-explosion plant, flywheel, accelerator, jointed drive shafts, carbureter—these complications are a splendid tribute to the "persistent inventive genius" which created them.

The intelligence applied to the problems was of the highest order, save that it was short-sighted. Instead of trying to find an easier road to their goal, instead of remembering that fork which they had passed, the pioneers persisted on the path they had selected, with just enough success each year to urge them forward. And

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now it is perhaps not too much to say that they have penetrated so far that they find themselves in the midst of an ambiguity. Today some of those who are more courageous are beginning to think for the first time of the other road, and to give an occasional glance at the lone voyager who has come so much nearer giving to the public that performance which it has always demanded and is still demanding.

Stored Power an Advantage of Steam Cars

The secret of the performance of the real steam car lies in its stored power. We have in mind a certain type of steam car, once most conspicuous, which had none of it and which was abandoned for the internal-explosive type. With its flash boiler, it was necessary, as in a gas car, to create the power after the need for it had arisen. This involved control mechanism as delicate and complicated as in a gas car, including even a gearshift and a clutch.

Difficulties with Internal-Explosive Engines

It is highly probable, if those early automobile engineers could have seen the latest products of their most brilliant successors, with eight or twelve cylinders; with standardization so remote, even in this respect, that the oldest manufacturer of cars in America admits that twelve cylinders are necessary for a good performance, and the most successful claims that four are enough; with a product running over 3000 r.p.m. and the resultant lubrication problems; with the engine removed to the farthest possible point from the driving-wheels; with two or even four universal-joints; with incredibly delicate vaporizing mechanism; with incredibly accurate manufacturing limits; with even four valves to a cylinder controlled by incredibly accurate timing gears; with the primitive functions of clutch and gearshift still present—it is highly probable that those early engineers, if they could have visualized the modern automobile, would have cried to one another, "Hold! We dare not impose so prodigious a tax upon the brains of engineers. We dare not impose so prodigious a mechanism upon the civilization we are striving to benefit."

But entirely aside from the complications of these later days, entirely aside from the initial and well-nigh prohibitive practice of making a firebox out of the cylinder, the error was fundamental.

There is no stored power in an internal-explosive auto-

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mobile—there cannot possibly be any. The only place where power can be stored is in the flywheel, and it can be stored there only by speeding up the engine after the emergency calling for the power has arisen.

Only in the steam car and in the electric car can power be stored.

The electric car would be ideal, and would probably displace both steam and internal-explosive cars but for its four principal limiting factors—its range of operation is too short; it takes hours to restore its power; it lacks a forty or fifty-mile-an-hour spurt; and the driver is conscious every minute that he has less power than he had a minute before.

Stored power is necessary for any self-propelled vehicle, on rail or road, for the performance the public wants and must have.

The modern internal-explosive engine is a marvel of ingenuity. "Persistent inventive genius" has done wonders in combating the difficulties which are inherent in it.

But no matter how well the work is done or shall be done, the results the engineers are striving for, the performance the public is demanding and must have, can never be attained with this form of power.

Attributes of the Ideal Car

We want to make, we want to sell and we want to drive a car

Whose whole range of power is controlled by a little finger lever on the steering-wheel; whose gears are always in mesh, so that we never have to separate the power from the load in order to change the car speed.

Whose power is built up in advance, ready for instantaneous application to the rear wheels at our merest wish.

Whose power does not depend upon the speed of a flywheel, which can be accelerated only after the critical emergency has arisen.

Whose maximum torque can be applied to the wheels under the most adverse conditions, giving fullest efforts at lowest speeds—at one mile an hour if the hill or mud requires it—without relation to the speed of the engine.

Which has only two cylinders, and a crankshaft only 8 in. long, and yet has 100 per cent torque on the crankshaft continuously, without "impulses"—without even "overlapping impulses."

Whose crankshaft is parallel to the rear axle so that no right angles must be turned by the power.

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Whose engine is at the nearest possible point to the driving axle—geared to it, in fact, so that no jointed drive-shafts are necessary.

Whose power is not created by turning the engine cylinder into a firebox, but is generated with no moving parts to be exposed to the heat and the gases.

Whose power can be built up without operating the engine, or whose engine can deliver this power without waiting for combustion.

Whose fuel is supplied for consumption without the aid of a carbureter or other outside atomizing or vaporizing device.

Whose fuel is kerosene.

If the engineers of the industry accept these as the attributes of the car they want to make, to sell and to drive, then we will accept the responsibility of affirming that they can be had only with steam.

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MID-WEST SECTION PAPERS

MAGNETO IGNITION FOR FARM TRACTORS

By J G ZIMMERMAN*

Many of the tractor products now being marketed show that not enough care is being exercised in installing ignition systems. The tractor engine has conditions to meet different from those of the automobile engine, and as a consequence, certain requirements must be more positively met. The object of this paper is to bring out several matters of importance pertaining to the proper installation of magnetos and to the principles underlying their construction.

In the tractor engine we have what is known as constant duty work—the engine is worked hard and at high load-factor most of the time; consequently, all appliances and accessories are subjected to harder service than in the automobile. This means then that such accessories must be well constructed mechanically and made so that proper attention may be given to keep them in order. It means also that the appliances must be conveniently located and well protected.

Not long ago a tractor manufacturer pointed out with pride his method of fully enclosing an engine and gearbox. I noticed with particular interest that the only unprotected part of his equipment was the magneto. It was placed where a single accidental blow could readily put the entire outfit out of commission. It was also placed where dust and dirt could collect in quantity. This particular tractor is now provided with better protection for the magneto, in accordance with suggestions I made.

Upon examining the installation in a truck engine, I found the magneto crowded into a corner under the dash with the breaker-box toward the rear. It was almost impossible to get one's hand through the available space to get at the breaker-box and absolutely impossible to adjust anything without first completely removing the magneto itself. Such an installation leads to poor service.

Another case in point was a tractor engine with the magneto so placed that one would have to put his arm

*Service Manager, Sumter Electrical Company

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through a loop in the exhaust pipe to touch it. Because of this location the magneto became very hot.

In still another four-cylinder tractor engine the spark-plugs were so placed as to run extremely hot. One of these engines, in use at a tractor school, was running on three

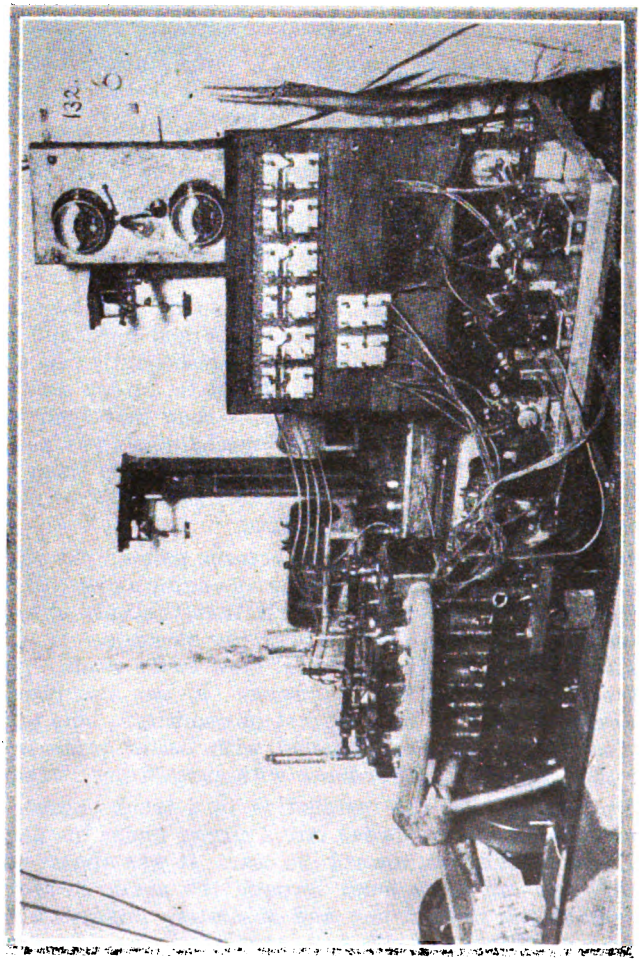

FIG. 1—APPARATUS USED IN TESTING SIX MAKES OF MAGNETOS. ALL DRIVEN BY THE SAME CAMSHAFT

cylinders. The students and instructors preferred to let it run on three rather than fix the faulty spark-plug. As ten or more makes of tractors were in use, the showing made for this particular one was poor.

In another instance the high-tension wire was sent out by the magneto manufacturer made up in harness or

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of proper lengths to suit the installation. The person who applied it used the shortest cable for the longest reach and the longest for the shortest, with the result that the short cables rested tightly on one cylinder head, while the rest fell loosely over the engine.

Principles Involved in Wiring

The wiring of any ignition system has a great deal to do with its performance. Poor wiring can put the best source of ignition out of synchronism with the engine. A case of this kind was found at a marine engine company's plant. Here the magneto was all right but part of the wiring ran through a metal tube for 4 feet or more. The tube being grounded and the wire inside it, the energy generated in the tube and thus lost was sufficient to make the sparks fail and also to be late when they did pass across the gaps.

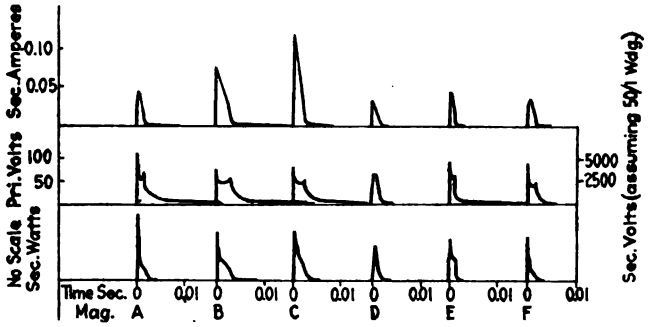

Fig. 2—Output Wave Forms of Various Types of Magnetos
Magneto speed 300 r.p.m. Watts are compared on the basis of like primary-secondary turn ratios

The remedy is obvious—do not use metal tubes to carry the high-tension wires. It takes time to charge long wires, consequently it is well to use the shortest possible wires of equal length from the ignition device to the spark-plugs. A wire placed too near the engine has its capacity to become charged increased. Hence all wires should be kept at least 1 in. from grounded or other metallic parts. Likewise, all wires should be kept at least ½ in. apart and should not all be run in the same tube, even though it be non-metallic.

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Spark Requirements

In order to understand the importance of proper wiring, let us next consider the character of a spark. Inasmuch as the period of inflammation of a gas charge is short (at 1200 r.p.m., ignition 30 deg. early, exhaust valves open 40 deg. before lower dead-center, it is 0.0236 sec.), the instant of ignition is important. It must be definite and not subject to variation; it must occur exactly when wanted and ignition should occur at a time definitely relative thereto, that is, the inflammation should start at some exact positive period after the spark starts. Hence, all ignition systems must be so designed that this will be effected. Fig. 2 shows a set of spark characteristics taken from several different makes of magnetos for rough comparison. Owing to the difficulty of measuring the spark (secondary) voltage of high-tension apparatus, we have taken advantage of the transformer relation of primary to secondary currents. If we measure the primary voltage and secondary current, we can, knowing the ratio of turns, approximate the secondary circuit characteristics. We hope, however, sooner or later to obtain the true secondary voltage characteristics; we are working along this line with some prospect of success.

From Fig. 2 we see that all types of magnetos act about the same way, save for the difference in values of voltage and current. The watts calculated from the other data are not expressed in relative terms, and, although one magneto appears to outclass another, it may not do so in actual performance, since no definite data as to winding ratios are at hand. The curves are intended to illustrate only the above-mentioned facts.

In Fig. 3 is shown a set of oscillograph records, which illustrate an important fact. The records are those of a magneto running at constant speed and having the secondary spark gap of needle points in air varied from short circuit to a gap too large for the voltage to jump. The secondary currents gradually increase as the gap is reduced from the open-circuit to short-circuit position; at the latter the current, instead of following the jump spark, is generated as the rotor changes the flux through the coil.

At open circuit, the current being zero, there is no energy available at the gap; likewise at short circuit there is none available there. Hence somewhere between the two extremes there is a gap where maximum energy will be developed.

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In this particular case the maximum is close to 0.25 in. As this test is made in the open air, at one atmosphere, and, since with a given voltage, the gap that will be jumped decreases as the absolute pressure is increased, the gap corresponding to the 0.25 in. between needle points in air will be one-sixth of 0.25 or 0.04 in. at 90 lb. compression pressure, or about six atmospheres. Now, since the voltage required to send a spark a given distance through air increases when the spark gap terminals are changed from sharp points or edges to rounded surfaces, the 0.04 in. is further reduced, say, to 0.03 in.

This explains why manufacturers recommend a gap of from 20 to 30 thousandths of an inch for their magnetos, as the best results are then secured. All magnetos are intentionally designed to generate about the same voltage under like conditions, although not always made to do so.

Initial Kick Voltage

Particular notice should be taken of the first voltage kick of the primary, which corresponds well with that of the secondary. This is true since the voltage developed by coils of wire over the same core must of necessity develop the same voltage per turn. This initial kick starts the spark. Needless to say, anything which chokes or holds back this kick lowers the efficiency of the spark. Hence the importance of the foregoing information regarding wiring; its influence on the spark effectiveness is apparent. Note that the voltage kick is almost instantaneous. This means that it is decidedly preferable to have the charging of the high-tension cable circuit instantaneous. Long cables, metal tubes for guiding them, cables near ground or each other, and hot insulation are all undesirable.

Tests made with a four-cylinder engine running on kerosene showed that most magnetos on the market meet the requirements of good ignition. Fig. 1 represents a test of six makes of magnetos driven by the same camshaft. Each magneto was given its own advantageous spark advance and in the test not one magneto developed the slightest difference in output. Any magneto could be thrown on at any moment, so that during the test the fuel, carbureter, load, spark-plugs, cables and water temperature were identical.

It was particularly noted on examining the timing that all systems employ almost exactly the same advance, that

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is, the first part of the spark does the work. With the magnetos delivering large or long-lasting sparks the charge was fired at the beginning before the whole spark was developed—proving that the follow-up was not necessary.

Magneto Make-and-Break Sparks

It must not be construed from the above remarks about the concentrated follow-up current, that a small follow-up will always suffice. In the use of low-grade

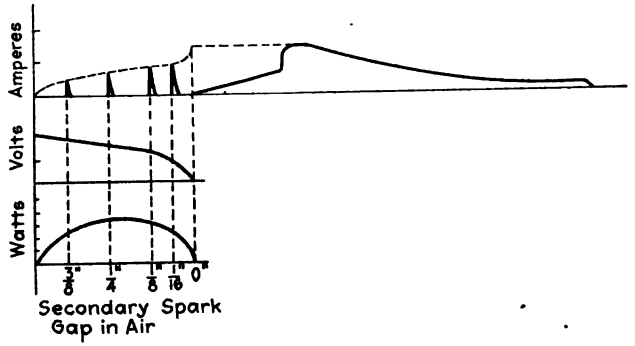

Fig. 3—Oscillograph Records of Secondary Currents and Volts with Variable Spark Gap

Maximum energy at gap of 0.25 in. in air, or about 0.04 in. under compression at 90 lb. per sq. in.

fuels, a spark of longer duration will undoubtedly help to insure good ignition although it may be perhaps a little irregular. Hence, low-speed engines using fuel oil or low-grade kerosene are preferably ignited by a "fat" spark. The make-and-break ignition system is here preferable because the energy of the spark is greater than in high-tension systems, and the fouling of a plug or igniter is less apt to occur. In the high-tension systems the voltage over the insulation is from 2500 to 4000 volts or more, depending on the compression temperature and spark gap, while in make-and-break systems it seldom exceeds 250 volts.

Fig. 4 shows the spark energy of a small "plug oscillator," which is a combination in one unit of a magneto and make-and-break mechanism, for one-cylinder engines only. The spark output of this is much greater than that of the largest standard high-tension magneto in use today. Now that we are forced to use low-grade fuels there is no reason why the make-and-break system should not prove as satisfactory as high-tension ignition on low-

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speed, single-cylinder traction engines. In the case of small farm engines, "plug oscillators" are proving a great advance over the older battery systems. One reason lies in the efficiency and mechanical durability. Another is the fact that with such a device the spark may be readily

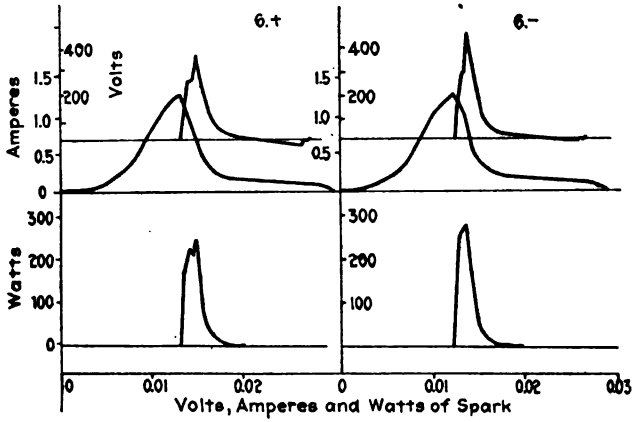

FIG. 4—SPARK ENERGY OF SMALL "PLUG OSCILLATOR"

tested at any time to prove its condition. It can also be cleaned readily. It is possible, therefore, that low-tension ignition may come into more extended use for tractors with single-cylinder low-speed engines.

HIGH-TENSION VOLTAGE AND STARTING

It is well known that the magneto develops a weaker spark with reduced speed and that at very low speeds (below 50 r.p.m.) it is difficult to start an engine on the magneto. As tractor engines are usually heavy and hard to crank, starting is difficult without some means to help to give a satisfactory starting spark. In the battery and coil system a separate battery is required, as is also a separate high-tension vibrator coil or a magneto coil with a master vibrator.

Fig. 5 shows the battery-coil system applied to a magneto. The sparks are all satisfactory. The user must remember to turn the switch over to "straight magneto" when the engine comes up to speed. Unless some extra automatic device is used with it, he usually forgets and finds that the next time he wants to start, the battery is "dead."

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Starter Coupling with Magneto

The starter coupling is a simple mechanical self-contained automatic or semi automatic attachment to a magneto by means of which the operator, after priming the engine, can crank it at any speed desired, when it will deliver a fine spark. After the engine is started, the device is cut out and the engine run on the magneto alone. The semi automatic outfit is preferable because it is seldom necessary to run a tractor engine below 100 r.p.m. For mechanical reasons the device is not suitable for continuous service, its object being to start the engine.

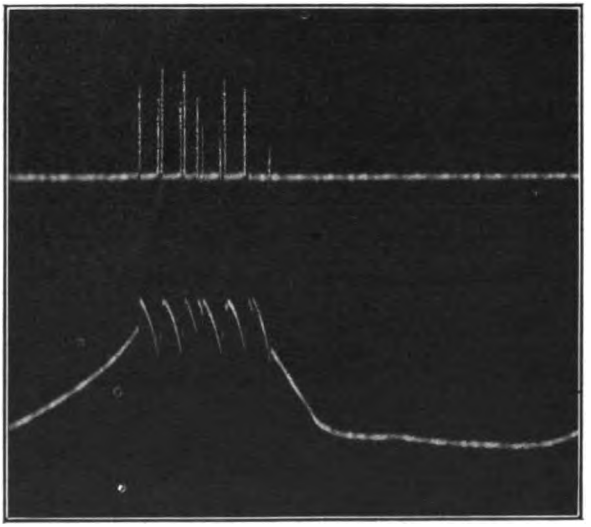

Fig. 5—Results with Battery-Coil System Applied to a Magneto

over straight rotary action. Here with the magneto operated at 120 r.p.m. the spark energy is less than half that obtained with the coupling. Any properly designed starter coupling should retard the spark, no matter where the advance lever may be located. The starter coupling, if used, insures greater safety to the operator. Fig. 7 shows the action of a starter coupling from start to throw out.

Location of Magnetos

Several tractor manufacturers place the magneto not only in an inaccessible, but also in a hot place. As a general rule the higher the temperature the poorer an electrical insulator becomes. Hence, the effect of heat

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on the insulation of a magneto is to reduce its efficiency. In the future a determined effort should be made to consider the magneto and its location.

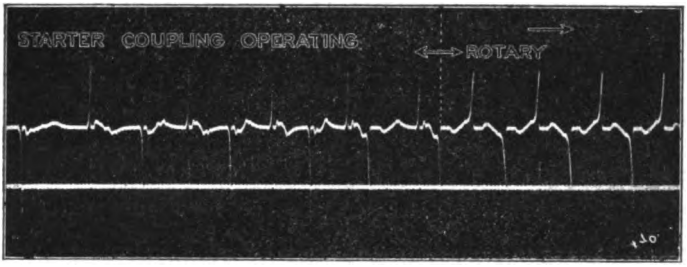

FIG. 7—RECORD OF OPERATION WITH A STARTER COUPLING

In conclusion I wish, therefore, to make a plea for giving the matter of ignition equipment more attention. Most complaints that come to the manufacturers result from either improper installation, making it impossible for the user to give proper attention; or improper attention, owing to lack of information on the part of the user.

Fig. 6 shows the advantage of the starter coupling

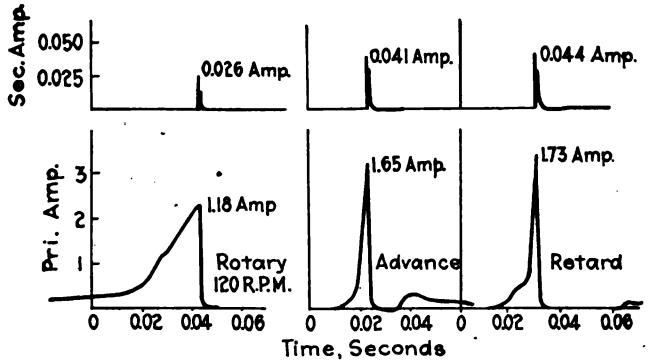

FIG. 6—RECORDS SHOWING THE MAGNETO OPERATION WITH STARTER COUPLING AND WITH ROTARY ACTION

THE DISCUSSION

DARWIN S. HATCH (M. S. A. E.):—What difference in the spark is caused by change of polarity?

MR. ZIMMERMAN:—The material composing the spark arc comes from the electrode and is converted into a vapor by the heat from the electric current. Although it is sometimes thought that this vapor is furnished by the negative electrode, it is in reality furnished by the posi-

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tive. This fact is evident when a moving electrode furnishes the arc material, as the characteristics of the spark, when the hotter electrode is rotated from the stationary electrode point, differ from those of a spark produced with the cooler electrode as the rotating member. In the latter case the vapor cannot be produced as rapidly.

In airplane engines operating at high speeds, the cooling of the spark-plug is a very important factor, for if the spark energy is great the point may become hot enough to preignite the fuel.

O. W. A. OETTING (M. S. A. E.):—It is true of magnetos as of all other electrical equipment that they are thought of last. The sooner engineers agree upon the correct application of electrical equipment to tractors and trucks, the better it will be for all concerned. Most automobiles now employ the battery system, although some still use the dual system, starting on the battery.

CHAIRMAN SMITH:—I recall a case where metallic tubes eliminated difficulty that had been experienced with fiber tubes.

MR. ZIMMERMAN:—It may have been caused by the fact that cables for all cylinders were bunched closely together, thereby causing the induction of static currents in the cables not in use and the static ignition of a charge in one or more cylinders, due to the discharge to ground of this static charge over the gaps of the plugs.

Note that a piston when beginning to act has a charge under low compression, though ignitable, and that the voltage required to jump the small gap of the plug in this cylinder is far less than that required under full compression. Hence the static charge may ignite this compressing cylinder and cause the trouble.

Now with a metallic tube substituted and this tube *grounded* the static charge in the wires is prevented because the tube absorbs the energy and conducts it to ground. It is safe to say that in such a case as this the energy of the sparks after installing the tube of metal would be less than if each cable were separately inducted to its plug and kept away from grounded parts on the other cables.

A MEMBER:—What is the ideal spark point?

MR. ZIMMERMAN:—I wrote a paper on that subject for the January, 1915, issue of the *Gas Review* outlining the characteristics of different spark-plugs and the type I recommended at that time as the ideal spark-plug has two concentric terminals with sharp edges. That type is now being employed in airplanes.

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ECONOMIC SIZE OF FARM TRACTOR

By ERNEST GOLDBERGER*

A couple of years ago a study of the cost of plowing with tractors would not have been of any practical value. The conclusions, even though they might have been drawn in the most logical way, would have been theoretical, because based upon theoretical assumptions.

During the last few years the line of tractors has been extended until now we have on the market practically any size of tractor, ranging from 10 up to 130 hp.; knowing their performances and prices, the requirements on the farm and the manufacturing problems—that is, all the premises—it is possible not only to draw correct conclusions as to present conditions, but also, what from a business standpoint is more important, to anticipate the future development.

Many students of this problem will undoubtedly say conditions on the farms are so different as to the quality of soil, size of farm, kind of crop and hundreds of other things, that, everything considered, a general and intelligent solution of the problem is precluded, while without taking everything into consideration the solution cannot be correct. That is, at least, as it appeared to me a couple of years ago, but I have changed my mind in the meantime.

One way of attacking the problem is by "national referendum"; that is, by having every tractor owner state the size of his tractor and farm, perhaps also the size he thinks would suit him better, and draw the conclusion according to the law of averages. The Department of Agriculture has published the results of such an investigation, directed by A. P. Yerkes, and although based on different methods the results of the survey agree with the conclusions that follow the thought of this study.

In the following I shall consider three sizes of tractor that would be built along exactly the same standards of quality, design and workmanship:

A—Capable of pulling two 14-in. plows, 6 in. deep, in average soil, at a speed of 2½ m.p.h. This would require about 9 hp. on the drawbar.

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B—Same quality tractor, of double size, pulling four 14-in. plows, 6 in. deep, in the same soil, 2½ m.p.h. speed. Drawbar 18 hp.

C—Twice as strong as B, pulling eight 14-in. plows, 6 in. deep in the same soil. Drawbar 36 hp.

Estimated Costs

Since there is today no doubt in the mind of anybody familiar with the tractor problem that really successful tractors are and must be built according to the standards of quality adopted by truck manufacturers, I shall estimate the prices as follows:

Initial Costs

Tractor A, $600; Tractor B, $1,500; Tractor C, $3,600. The price for Tractor A, if the design, material and workmanship be first class, cannot be reached unless quantity-production methods are adopted. For 100 per cent increase in size the price increases 150 per cent in my estimates, since I make the preliminary assumption that with the size of the tractor increasing the demand might decrease and the manufacturing methods to be adopted become more expensive.

I shall not consider the cost of plows and other equipment, since in a comparative study of the cost of plowing with different sizes of tractors this matters very little.

Cost for Operator

A two and a four-bottom tractor and plow can easily be operated by one man, while for an eight-bottom tractor two men are necessary; not for lifting and lowering the bottoms, which can be done automatically, but for cleaning and unclogging the plows, which would mean stopping the machine often if this work were to be done by the man steering the tractor.

The expense for operators would therefore come to 30 cents per hour for the two and four-plow outfits, and 60 cents for the eight-plow outfit.

Fuel Cost

Experience has shown that the fuel consumption of gasoline tractors per horsepower hour changes little or none with the size of engine, provided all are built ac-

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cording to the same standards and are run at the same percentage of load. In our case, the cost for fuel (and oil, etc.) should be, for tractor A, 25 cents per hour, tractor B 50 cents per hour, and for tractor C $1 per hour.

Depreciation

One often hears the statement that a tractor will last five or six years. Such a statement is fundamentally wrong. A tractor, or any machine in general, depreciates because of, first, the wear and tear of the machine, a factor altogether independent of the age of the machine, the only thing affecting it being the actual working hours; and second, the obsolescence, which, on the contrary, depends upon the age of the machine and not upon the working hours. The latter factor is more important than the first in automobiles or any products subject to change of style, but with utility products it has little importance, unless a new invention changes fundamentally the existing conditions. Yet, even among automobiles, the Ford car, model T, has not become obsolete after ten years of use; and I know of American tractor manufacturers who have not improved their tractors materially since 1911-12, and are now selling more tractors of the same type in a month than they made all told up to 1915, while some of the machines they sold in 1912 are still "going strong." Therefore, it is appropriate to base the allowance for depreciation upon the total actual working hours of a tractor, provided it be of progressive design, and not upon total life. This should apply satisfactorily to horses, or any draft animals whose average life is nearly constant whether they work or not. It is a conservative estimate to say that tractors of modern design, of any size, will be good for only about 4500 hours of plowing; this would give a tractor, plowing 10 hours a day, a life of 5 years. Should the tractor plow only 45 days per year its life would then be 10 years, provided no other work but plowing is done. This estimate is safe, since 80 per cent of its value has already been figured in the item for wear and tear, and part of the remaining 20 per cent can easily be recovered by the sale of its high-grade material as junk.

For tractor A the depreciation per plowing hour can be figured at 600/4500 = $0.13; for tractor B the figure is 1500/4500 = $0.33 per plowing hour; for tractor C at 3600/4500 = $0.80 per plowing hour.

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Repairs and Interest

These two factors are considered together, and estimated at 6 per cent per year on the initial price of the tractor. Since the value of the tractor decreases constantly, the interest on this value also decreases. If it is assumed that the tractor will last 6 years, the interest during the first year will be 6 per cent on the price paid; the following year it will be 6 per cent on the then value of the tractor, or 5 per cent of the total initial

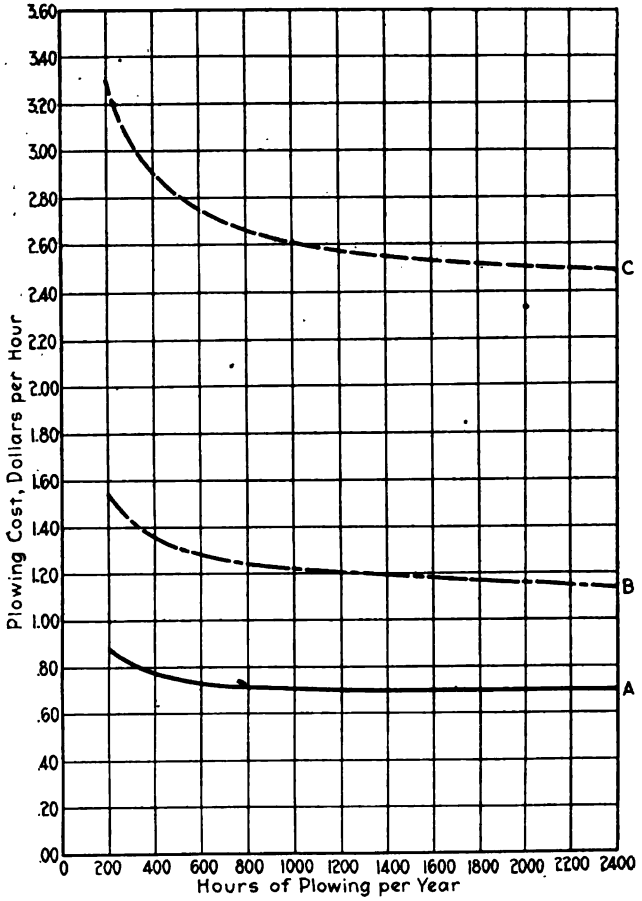

FIG. 1—APPROXIMATE COST OF PLOWING PER HOUR FOR THREE SIZES OF TRACTORS

A, two-plow; B, four-plow, and C, eight-plow

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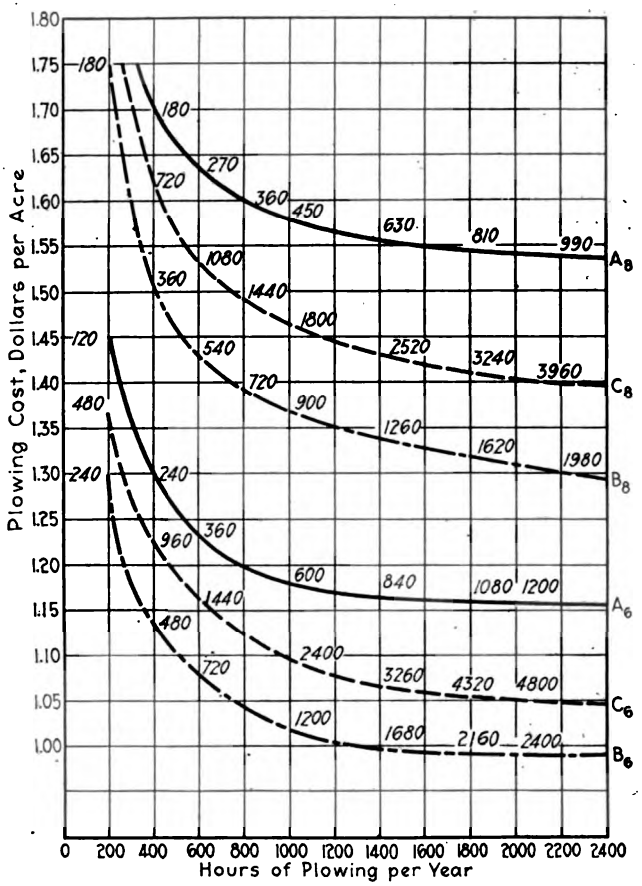

FIG. 2—APPROXIMATE COST OF PLOWING 6 AND 8 IN. DEEP WITH THREE SIZES OF TRACTORS

A, two-plow; B, four-plow; C, eight-plow. Figures above curves show number of acres plowed in corresponding time

price; and so on. The allowance for interest and repairs being 6 per cent per year on the total initial price, the result is that no repair allowance is made for the first year, 1 per cent of total initial value is taken for the second, 2 per cent for the third, and so on; such an estimate corresponds with the actual conditions as they exist with tractors of modern design.

COMPARISON OF PLOWING

Suppose we figure every one of these items for the tractors chosen for a variable number of actual plowing

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hours per year; in Fig. 1 the abscissas represent the number of actual plowing hours per year, and ordinates the cost of plowing per hour; the curves marked *A*, *B* and *C* correspond to the machines above mentioned.

Fig. 1 does not give a final indication as to the comparative cost of plowing with the different sizes of tractors, but is used in making Figs. 2 and 3. Fig. 2 gives the relation of cost in dollars per acre to the number of hours of plowing. Tractors A, B and C are supposed to pull 2, 4 and 8 bottoms respectively, 14 in. wide, 6 in. deep, at a speed of 2.5 m.p.h. It is assumed that the tractors will plow only 85 per cent of the time they

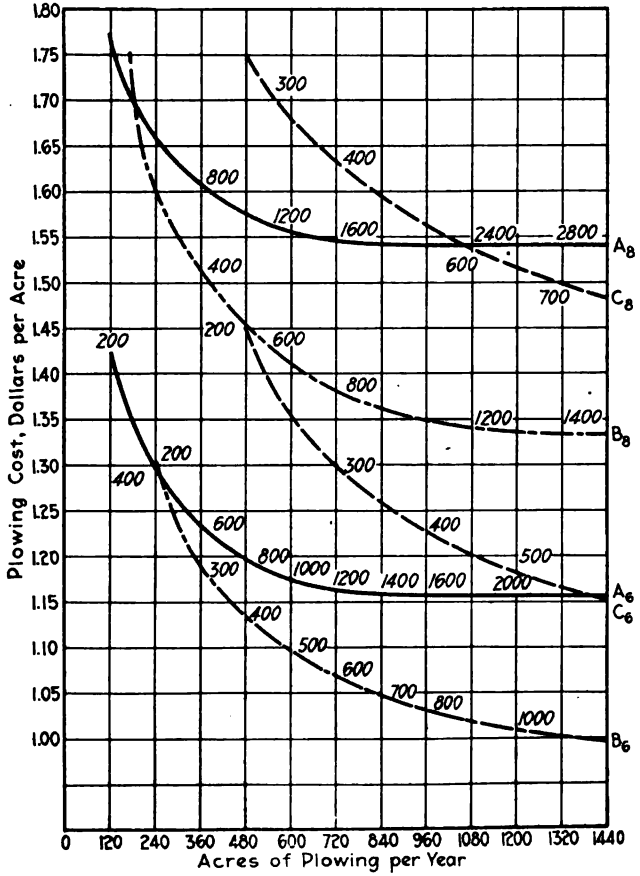

FIG. 3—CURVES FOR SELECTING ECONOMIC-SIZED TRACTOR FOR ANY NUMBER OF ACRES, WITH TIME REQUIRED AND COST

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are in the field, 15 per cent being the allowance for turning the headlands, oiling, trouble, etc. The area actually plowed is calculated as 0.6, 1.2, or 2.4 acres per hour respectively. The small figures along the curves represent the number of acres per year the chosen tractor would actually plow.

From Fig. 2 as a basis, Fig. 3 has been drawn, and this gives the final information we are after. We read the area to be plowed in a year on the abscissa, follow the ordinate until it intersects the curve representing the A, B and C tractor, and then at the left we read on the ordinate the cost of plowing per acre.

If we have to plow less than 250 acres per year, and not deeper than 6 in., the tractor with 9 db.hp. plows the cheapest; for anything above this amount, the four-plow tractor with about 18 db. hp. is the cheapest in the end; while even for farms so large that one four-plow is too small, it pays better to have two four-plow tractors than one eight-plow tractor. The advantage of having two smaller powerplant units instead of one large one, or the matter of flexibility, is not considered at all in the charts.

Cost of Plowing

The chart shows much more than this. It shows how much cheaper the plowing is if there be much of it to do, or in other words how expensive it is to be poor. There are, however, two ways by which the small farmer can compete with the big one without having to buy his neighbor's land: by deeper plowing and by plowing early and quickly. To meet both of these conditions the two-plow tractor will give way to the larger one; in fact, from the upper curves, representing the cost for plowing 8 in. deep, we find that for above 180 acres of plowing per year the four-plow tractor is the cheapest. For 240 acres of plowing per year the four-plow tractor is only a little cheaper in dollars and cents, but it has the advantage that it performs the work in about 30 days instead of the 60 necessary with the smaller one.

Increasing the Depth of Plowing

Now, there are two ways of making a tractor intended to pull four bottoms 6 in. deep plow 8 in. deep. First, by hitching three plow bottoms instead of four; second, by leaving the four on and plowing at a speed of about 1.9 m.p.h., which every tractor should have, instead of at 2.5 m.p.h.; the area plowed per hour and the cost are

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then nearly the same. In fact, there is a great advantage by applying the first method, since we cannot count on using the full power of the tractor continually where its efficiency is highest. Hard spots on the land, or grades, will be overcome by shifting to the low-speed gear. On the other hand, with a two-plow tractor, intended for plowing 6 in. deep, we have no other choice, if extra expense for specially narrow bottoms is to be avoided, but to run the plows 8 in. deep on low speed all the time.

Author's Conclusion

Manufacturers have too long likened the tractor problems to those of truck manufacturing. There is good reason for building four or five sizes of trucks in the same plant. Sometimes a 5-ton load cannot be handled by either one or two 3-ton trucks, nor is it economical to run a 3-ton truck with a 1-ton load; but in plowing the load can be easily adapted to the full power of the tractor.

It is important to note that a conclusion has been reached in favor of the four-plow tractor, although the preliminary assumption was made that the two-plow tractor will probably be the more popular type, and therefore the cheaper to manufacture. But since we have reached the conclusion that the four-plow outfit is the more economical it can be expected, owing to its production in large quantities, that its price per horsepower will not be 25 per cent higher than that of the two-plow outfit, as assumed at the beginning of this analysis, but perhaps even lower.

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MINNEAPOLIS SECTION PAPERS

ROLLING RESISTANCE OF TRACTOR WHEELS

By AMOS F MOYER*

Physicists define a plastic body as one that can be molded under pressure, recognizing that the rate at which deformation is produced often affects materially the plasticity of the substance. Funk & Wagnalls' New Standard Dictionary states that the body must permit "molding of form," which might lead to the assumption that the volume is to remain unaltered.

The freshly tilled, sandy, black loam soil used for the experiments described in this paper is not only capable of molding in form but is to a slight degree compressible as well. The yielding properties of the soil, however, appear to be due to lateral movement under vertical pressure to a much greater extent than to the compression of any given amount into smaller volume.

The mathematical derivations which it was desired to verify by these experiments are based on the assumption that soft soil, and probably other plastic materials, bear up a unit pressure proportionate to the depth of depression. This is equivalent to saying that the total weight borne up is proportionate to the volume perpendicularly inclosed by the displacing surface in actual contact with the substance and below the surface from which the substance has been displaced. In the case of a hard cylindrical body rolling on a level plastic surface, this volume consists of the prismatic half segment of the cylinder below the level surface and on the forward side, as illustrated in Fig. 1.

If the deforming body, instead of being a cylinder, were rounded upward in front and continued flat for some distance backward, so as to rest in contact with the bottom of the track, like a sled runner or tracklayer tread, the supporting volume of displacement would include also the rectangular prism extending backward a distance equal to the flat supporting surface. This illustrates the much greater carrying capacity which characterizes such surfaces.

Throughout the detailed studies and experiments thus far conducted, attention has been confined to plain flat-

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tired wheels of varying weight, width and diameter. Since the parabolic area (see Fig. 1) is equal to 2/3 *ad* the law already outlined for a cylindrical body or plain flat-tired wheel is expressed as follows:

$$W = \tfrac{2}{3}\, ad\, Fn \qquad (1)$$

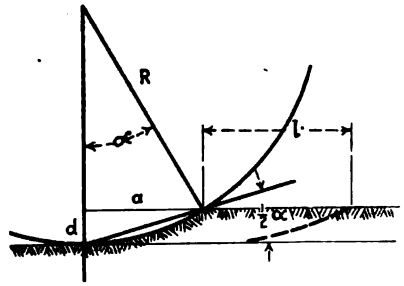

FIG. 1—ILLUSTRATING DEPRESSION OF SOIL BY TRACTOR WHEEL

where W = weight on wheel, pounds;

F = width of tire face, inches;

n = bearing value of soil, pounds per cubic inch.

But since $a = R \sin \alpha$ and $d = R - R \cos \alpha$,

$$W = \tfrac{2}{3}\, Fn\, (R^2 \sin \alpha - R^2 \sin \alpha \cos \alpha)$$

$$= \tfrac{2}{3}\, F\, R^2\, n\, (\sin \alpha - \tfrac{1}{2} \sin 2\alpha)$$

$$\frac{W}{\tfrac{2}{3}\, F\, R^2\, n} = \sin \alpha - \tfrac{1}{2} \sin 2\alpha \qquad (2)$$

which expresses a similar law in terms of the wheel dimensions and the arc α, which is in contact with the soil.

According to the principle of the conservation of energy, the work done in rolling a wheel a given distance must be equal to that expended in depressing the soil over the same distance. This fact, expressed mathematically,† results in

$$G = \frac{P}{W} = \tfrac{3}{4} \tan \tfrac{1}{2}\, \alpha \qquad (3)$$

Here P represents resistance or pull (in pounds) required to roll the wheel, and this divided by the weight on the wheel is G, or the equivalent grade. This is the percentage grade up which the wheel would have to roll

†The mean unit press. $= W/FR\sin\alpha = 2/3$ max. unit press. Therefore the max. unit press. $= 3/2\ W/FR\sin\alpha$ and varies as the depth from zero at the surface. The work per unit area $= d \times \tfrac{1}{2} \times 3/2\ W/FR\sin\alpha$ and the work of depression over distance l with resistance P is $dFl \times 3/4\ W/FR\sin\alpha$ which equals Pl. $P/W = 3/4\ d/R\sin\alpha = 3/4$ slope of chord $= 3/4 \tan \tfrac{1}{2}\alpha$

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without friction on a hard surface to give the horizontal resistance P. The term $\frac{3}{4}\tan\frac{1}{2}\alpha$, or three-fourths of the slope of the chord subtending the arc of contact, is a considerably smaller quantity than some of the catalogs of caterpillar tractors would have us believe for the loss in rolling a round wheel. One such catalog gives 41½ per cent equivalent grade for a 6-ft. wheel sunk in the soil 3 in., whereas the law just derived gives 15.6 per cent.

In the left-hand member of (2), the factors in the denominator other than n are evidently proportionate to the cylindrical volume of the wheel, which we may call Q. Throughout the calculations this has been expressed in cubic feet, so that W/Q, which is designated as S, or specific load, is written in pounds weight per cubic foot of wheel volume. The mathematical proof that the relation between S and G is fixed, whenever n remains constant, is as follows:

$$\frac{W}{\pi R^2 F} = \tfrac{2}{3}\pi(\sin\alpha - \tfrac{1}{2}\sin 2\alpha)n \text{ and } \frac{W}{Q} = S = f(\alpha)$$

From (2)

also

$$\frac{P}{W} = G = f'(\alpha) \qquad \text{From (3)}$$

Therefore when n is constant G has a fixed relation to S. I am indebted to Profs. W. E. Brooke and C. F. Shoop, each of whom made an independent solution eliminating α from equations (2) and (3) for the expression of the exact relation between these quantities, which is:

$$S = \frac{281{,}440G^2n}{(16G^2+9)^2}$$

$$\text{or } n = \frac{S(16G^2+9)^2}{281{,}440G^2} \qquad (4)$$

Experimentally it has been easy to determine S and G so that equation (4) has been very useful in studying the values of n.

It can be shown that $\cos\alpha = \sqrt{2P/FR^2n} - 1$ and $d = R - R\cos\alpha$, so that:

$$d = \sqrt{\frac{2P}{Fn}} \qquad (5)$$

$$P = \tfrac{1}{2}d^2Fn \qquad (6)$$

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This interesting deduction results in equation (5) expressing depth of depression and (6) expressing resistance. Such a simple expression as (6) seems almost uncanny when we realize that if we know enough about the soil, that is, if we know n we can tell the resistance of a wheel if we never have seen anything but its track.

Recording Dynamometer Built

Desiring to yerify some of these deductions, it seemed necessary to provide a course where conditions could be as quickly and as nearly duplicated as possible in order to obtain a large number of comparable readings. The only condition possible for quick duplication seemed to be freshly tilled soil, making possible the repeated use of one and the same bed. This condition at once suggested the use of a stationary dynamometer, which would measure the pull required to move loaded wheels by means of ropes. Knowing the value of averages for such readings, a recording dynamometer, Fig. 2, was designed and constructed.

Here the rope r is shown leading in from the left horizontally as in pulling a wheel. The sheave A is mounted on the vertical strut B, which in turn is pivoted opposite the lower sheave C in such a position that when C is driven at the required speed the rope in passing downward is parallel to B. This portion of the rope therefore produces no rocking reaction in B, while the horizontal portion exerts a force P, which is transmitted to the flexible tension member D. The right-hand end of D is wrapped over the top of the shaft E. This shaft terminates at both ends in small diameter trunnions which rest on the periphery of the large rollers F, thus greatly reducing the friction. Upon E is mounted the pendulum G, the horizontal displacement of which will be proportionate to the pull P in the rope.

On one extremity of E is mounted the crank-arm H, to which is pivoted the arm I. The lower extremity of I is attached to a link and constrained to move only in a vertical line, so that pencil J at the upper end of I describes a straight horizontal line after the manner of the Thompson indicator pencil motion. Since H is in line with pendulum G, the recording pencil J will move horizontally always in fixed proportion to the pull P. To damp the oscillations of the pendulum a loose-fitting oil dash-pot is provided as shown.

The recording paper feed is driven by the ascending rope passing over a large transverse sheave, which in

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turn is geared to the recording drum so that ½ in. on the record represents 1-ft. movement of the rope. Two interchangeable pendulum weights, one of lead and one

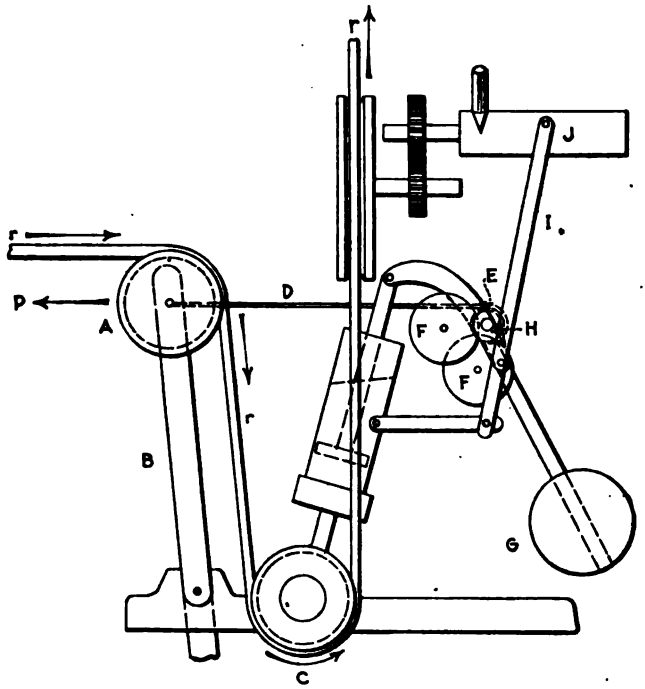

FIG. 2—DIAGRAMMATIC LAYOUT OF RECORDING DYNAMOMETER

of cast iron, have been employed, giving record scales of 80 and 40 lb. per inch respectively. With the lead weight the dynamometer will record pulls up to 250 lb.

Fig. 3 shows a typical record. The peak wave at the left-hand end is caused by the inertia effect at starting the wheel. A portion of the record is selected representing average conditions and the area determined by the planimeter over a measured length, thus establishing the average pull.

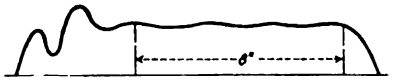

FIG. 3—TYPICAL DYNAMOMETER RECORD

The soil bed employed was at first 3½ ft. wide—later widened to 5 ft.—and 8 in. deep, of sandy black loam with only sufficient moisture to avoid dust and "ball"

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slightly when compressed in the hand. A pair of wheels were mounted rigidly on a shaft with two ropes leading to the dynamometer. These ropes were permitted to unwind from the shaft as the wheels advanced, thus eliminating bearing friction. Wheels were easily interchanged, and increments of load were obtained by fastening weights to the wheels in any balanced position.

Early Experiments

The deductions regarding G and S were suspected before they were proved, so that we early began seeking for the relation between the two quantities. While the studies have not yet resulted in mathematical expressions for all variables, we believe that the time devoted to them has been of great profit. On first thought it was not anticipated that n would be anything but constant for a given soil condition, so that S and G of equation (4) would always be plotted as a single line.

For a considerable number of the earlier tests these quantities did fall close to the same line, but at slow speed, in the vicinity of 20 ft. per min., with the heaviest wheels moving somewhat more slowly, there were unaccountable variations. The fact that the readings taken with the heavy wheels did not check with the rest of the readings led to the conclusion that velocity was the disturbing factor.

Before this conclusion was reached, however, some interesting studies were made on the law of three-quarters slope with relation to G as expressed in equation (3), and also on values of n determined from actual displacement.

Two wheels of 33¾-in. diameter and 4⅝-in. face, set 13½ in. between tires, were drawn through the freshly spaded and leveled soil bed at about 20 ft. per min. for light loads, but at lower velocities for heavy loads. An attempt was made at first to determine the depression by scaling the track after the wheels had passed over. By this method only a fair check of the law of three-quarters slope was obtained, but values of n calculated from displacement varied as much as 300 per cent. It became evident that the track was not the true level to which the wheel sank in passing over the soil. Next a series of 20 swinging stops was arranged above the course so that as the wheels passed beneath, each stop would be pushed up to and left at a point marking the position of the tire of the moving wheel. In this way it was proved that the bottom of the track is higher than

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the lowest point of the tire. The amount of this difference was measured as carefully as possible, taking into account several allowances for inaccuracies of hastily constructed apparatus, and from these data the gratifying check on the law of three-quarter slope expressed in equation (3) was obtained, ranging from 0.726 to 0.777 for the value of three-fourths in the formula. Values of n also became much more nearly constant, ranging from 4.28 to 6.27. Subsequently it became evident that this variation was probably caused by diminished velocity at the heavier loads.

Varying Velocities

Fig. 4 illustrates the amount of labor involved in making a comprehensive study of the effect of velocity on the resistance of a single size of wheel. For each point or observation it was necessary to spade up by hand the entire 3½ by 25-ft. course. Fig. 4 shows the resistances obtained for a pair of wheels, 24¼-in. diameter by 6¼-in. face, pulled at varying velocities with six increments of load. The lines show clearly that resistance increases with velocity for each weight on the wheels, and that this increase is somewhat more rapid with heavy loads than with light loads. All these lines when produced were found to intersect at the left in a common point representing zero resistance and a negative velocity of 1013 ft. per min. or minus 159.4 r.p.m. The full meaning

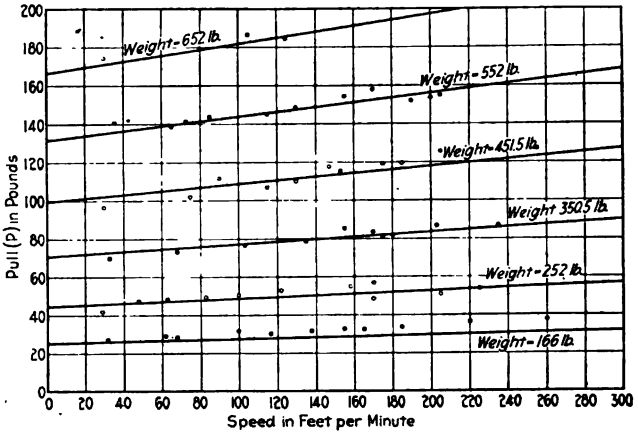

Fig. 4—Resistances for a Pair of 24¼ by 6¼-In. Wheels with Different Loads

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of this is difficult to comprehend, but the fact is nevertheless of value to us in making deductions.

By reading off the intersections on any ordinate representing a desired speed, values for P will be determined of greater accuracy than if the readings were taken at precisely that speed, since the variation of points from the line indicates the considerable error in a large proportion of the individual readings.

From the theoretical equation (4) giving n in terms of S and G, and the values determined from intersections, it was ascertained that the value of n remained virtually constant for each speed, but varied for different speeds. It was by use of the average value of n obtained from the other lines on this plate that the lower line was dropped below the points of observation, assuming that there had been some undetected error in these readings.

Fig. 5 represents results from the intersection readings at 205 ft. per min. or 2⅓ m.p.h. after the values have been converted to S and G, specific load and equivalent grade. The six points shown correspond to constant values of n, and the curve produced to the origin is obtained by substituting this same value in the theoretical equation. As the general form of this curve, which may be called a characteristic, suggests the parabola, a determination was made by trial of the parabola having the analytic form $y^2 = 2px + a$, which most nearly cor-

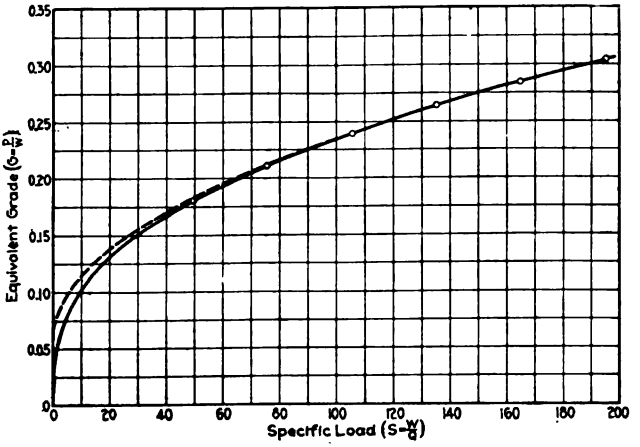

FIG. 5—RELATION OF S TO G AT 205 FT. PER MIN. SPEED (EQ. 4)
Dotted line shows relation of S to G, based on parabolic equation 7

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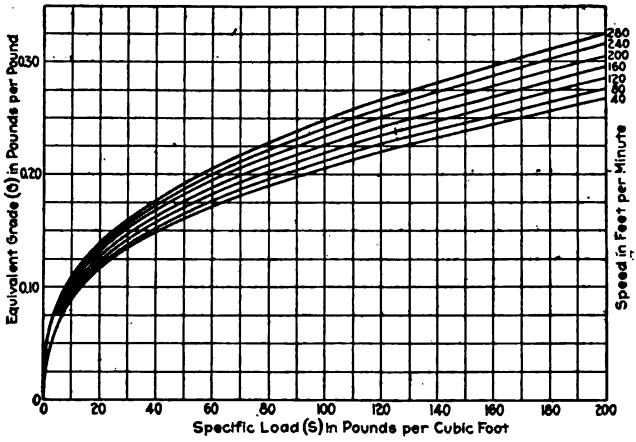

FIG. 6.—SHOWING RELATIONS OF S TO G AT VARIOUS SPEEDS

responds to the actual curve throughout the practical range of values. This parabola is shown by the dotted line, which coincides with the full line throughout the right-hand part of the field, and probably for some distance beyond. The equation of this parabola, which expresses G in terms of two constants and the square root of S, is as follows:

$$G = 0.0175\sqrt{S} + 0.060 \qquad (7)$$

Since S is weight per cubic foot of wheel volume, the equation may be rewritten with a constant divided by the diameter outside of the radical, and weight per inch of tire face under the radical, giving a simple formula for G in terms of weight and wheel dimensions. Unfortunately however this formula cannot be universally applied, as will be shown later.

When readings are taken at different ordinates representing increments in velocity, values of n will be obtained, each of which gives a different characteristic curve corresponding to the speed. Fig. 6 shows this series of curves determined from the same data obtained from the 24¼-in. by 6¼-in. wheels. Each of the lines represents a certain velocity corresponding to a different value of n. Since n to the first power is the common factor, the variation of which shifts the position of the curve, we may divide through by this and plot G against S/n when all the curves unite into a single line, Fig. 7.

Here are represented seven different velocities and

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six different weights, or forty-two possible values, all of which fall on the same line. In the theoretical equation

$$\frac{S}{n} = \frac{281{,}440G^3}{(16G^3 + 9)^2} \qquad (8)$$

it is evident that if the right-hand member can be approximately written as an exponential of G, it will be possible to express G in terms of S/n. To accomplish this the points were transferred to logarithmic plotting paper, giving the straight line shown in Fig. 8. This gives the following approximate exponential equation

$$G = 0.0588\left(\frac{S}{n}\right)^{0.286} \qquad (9)$$

which is universally applicable for values of S/n between 10 and 100, provided n is known for the conditions obtained. If S were expressed in pounds per cubic inch of wheel volume instead of pounds per cubic foot, the value of the constant 0.0588 would of course be altered. Since n, the displacement value of the soil, is already given in pounds per cubic inch, the parenthesis S/n would then become a mere ratio of specific wheel-load to displacement value. On ordinary plotting paper values of S/n between 100 and 300 have nearly a straight-line relation to G, as represented by the equation

$$G = 0.00140S/n + 0.210 \qquad (9a)$$

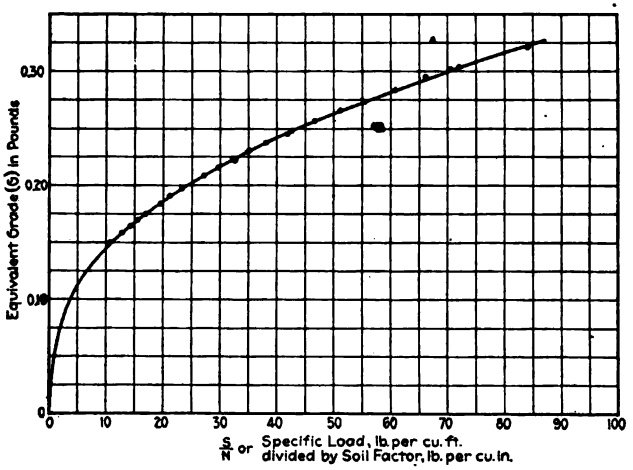

FIG. 7—RELATION OF $\frac{S}{n}$ TO G, BASED ON EQUATION 8

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The laws concerning the values of n are not yet completely ascertained. As shown in the data from this individual pair of wheels, n varies with the velocity. This relation was determined for the case and plotted in Fig. 9. The slight curvature and downward slope would indicate that n probably becomes zero at high or infinite velocities. On the other hand there were indications of zero resistance to rolling at a velocity of minus 1013 ft. per min., corresponding to infinite values of n, which was therefore plotted on logarithmic paper against 1013 plus velocity. Fig. 10 shows this to be another straight line in a position representing the following hyperbolic function equation:

$$n = \frac{74{,}990{,}000}{(1013 + V)^{2.413}} \qquad (10)$$

Varying Diameter

But this expression for n was determined from the same single pair of wheels, where one or more lines had been "doctored" to make n uniform for each speed. While a fair check has been obtained from other widths of the same diameter, other diameters show a consistent variation. A pair of larger diameter, 33¾ by 4⅝ in., not only gave different values of n at corresponding speeds, but at any given speed there was also a consistent increase in n with increments of load. This

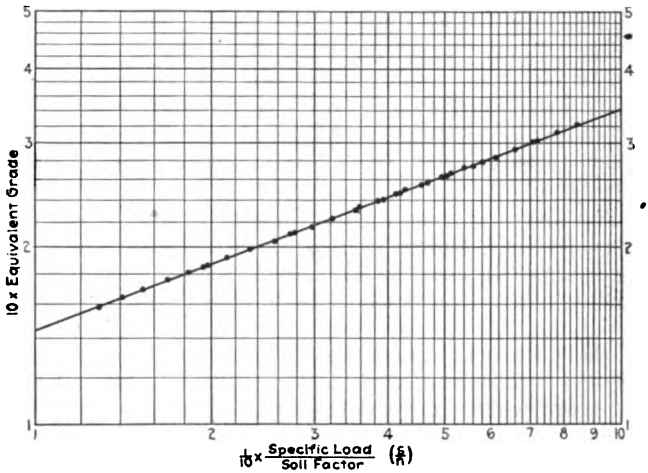

Fig. 8—Logarithmic Relation of $\frac{S}{n}$ to G (Equation 9)

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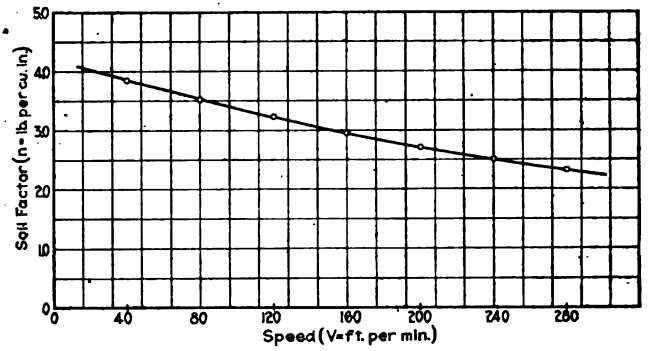

FIG. 9—SHOWING HOW BEARING VALUE OF SOIL VARIES WITH SPEED

seemed to indicate that the packing of the soil under a larger diameter of wheel was of such a nature as to cause a continually increasing amount to be borne up per cubic inch displaced with increasing loads. If this was true of a wheel larger than 24¼ in., would the reverse be true of a smaller wheel? Trial with a pair of wheels of only 15¼-in. diameter proved that with increasing loads decreasing values of n were obtained at any speed. This indicated another law as affecting the value of n, it appearing that only one diameter would give a constant n for each speed.

One or more lines on Fig. 4 did not coincide well with the points obtained in the tests on the 24¼-in. wheels, and it was found after making corrections that there was also a variation in this case in one respect similar to that found with both the other diameters. Instead of the lines representing constant n being true ordinates, as assumed in Fig. 4, they were found to be curved and concave to the left, being tangent to the vertical at about half their height.

The 15¼-in. wheels gave lines of constant n concave to the left but sloping backward, while the 33¾-in. wheels gave similarly concave lines, but sloping forward. Apparently larger diameters will give these lines more nearly flat, but curved in the same direction. The nature of the curves obtained are such as to render a mathematical representation difficult. Each line of constant n for large wheels is relatively flat at the left and curves upward more rapidly at the right. With all wheels of larger size than about 24 in., increments in diameter have not only the effect of increasing the volume of the

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wheel and reducing S but also of increasing the value of n, so that S/n is materially reduced, and hence also the value of G as expressed in equation (9). This argument then favors large diameter for load-carrying wheels.

As regards the variation in resistance with change in velocity, it is encouraging that wheels of different diameters show coherent results in the point of intersection of the produced resistance lines, as referred to on Fig. 4. This intersection occurs at zero resistance and minus 159.4 r.p.m. for all wheels, the value of 1013 ft. per min. applying only to the 24¼-in. diameter. It therefore appears that there remains one important law which it is not yet possible to write mathematically, namely, the variation of n with wheel diameter and depth of tillage. When this can be done it will be possible from the foregoing to express a general wheel-resistance formula involving the ordinary variables in a manner convenient for practical use.

After all of these determinations for the value of n are made, it may be asked whether the values actually represent weight borne up per cubic inch of soil displaced, or merely a hypothetical "something" which will satisfy a complicated equation. In obtaining data on the actual displacement of soil, there was no means at hand for ascertaining the "wave" action in front of the advancing wheel, merely the volumes being calculated which were below the original soil surface. Actually

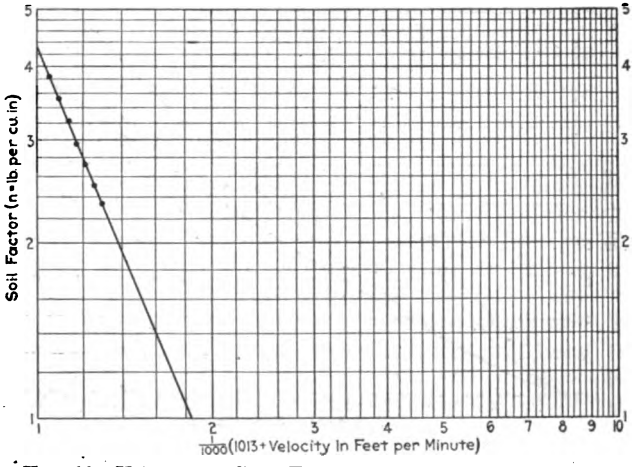

FIG. 10—VALUES OF SOIL FACTOR BASED ON EQUATION 10

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this wave action plus the movement of the small bank of loose earth which always forms in immediate contact with the tire effect a greater virtual displacement than would be obtained by the method last described and hence a smaller value of n. This is also what is obtained by the mathematical determination of n from S and G. Thus we see that the mathematical determination of n takes into account several variables which would be difficult to determine by measurement. When the laws of variation of n are once determined, the values thus established will give true values of resistance, since resistance is the quantity whence they were derived.

In many theoretical solutions of engineering problems, quantities are encountered which make difficult the mathematical expression of exact relations, and short cuts based on experiment are often expedient. Fig. 11 shows directly the experimental results from the three sizes of wheels previously discussed. From these curves of P against W at 2⅓ m.p.h. some interesting deductions can be made.

Using any desired increments in weight per inch of tire face, the total load W and resistance P can be determined, and hence also G, which equals P/W. These values independently determined for each wheel size are plotted against diameter in Fig. 12. Up to date we have only three points of observation on these curves and in several instances but two. It seems reasonable to expect,

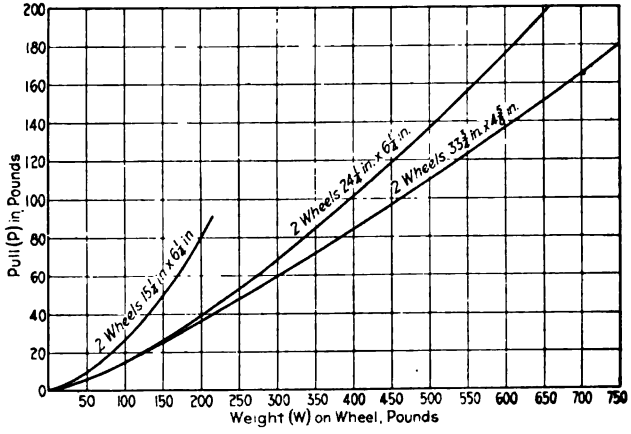

FIG. 11—RELATION OF W TO P FOR THREE SETS OF WHEELS AT SPEED OF 205 FT. PER MIN.

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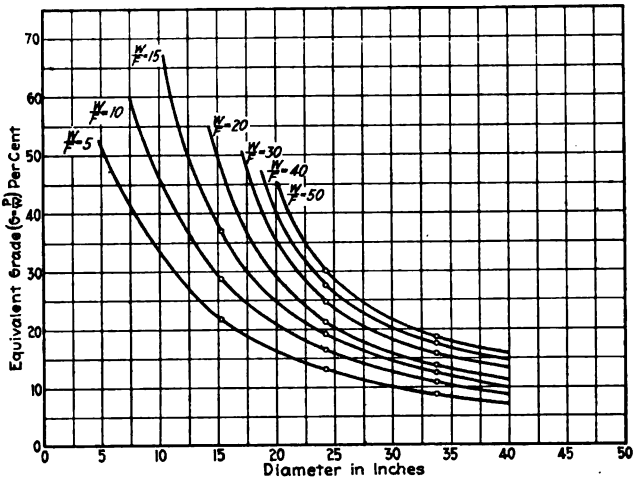

FIG. 12—RELATIONS OF EQUIVALENT GRADE AND WHEEL DIAMETERS
Speed 205 ft. per min. Various weights per inch of tire face (W/F)

however, that these lines are of a quasi-hyperbolic nature, as we would expect G to diminish with large diameters and to increase rapidly with absurdly small wheels if the weight per inch of tire face were kept constant. A chart of this kind when completely-determined over wide ranges of diameter and load will in many respects be as useful as a theoretically correct formula. Indeed, it may be found possible to develop an empirical formula representing approximately this series of lines. At the present time this possibility does not seem as remote as does the mathematical expression of n in terms of diameter and load.

AUTHOR'S CONCLUSION

All tests to date have been conducted on the same soil in approximately the same condition. There are unavoidable differences between readings, which necessitate averaging the results from several runs or taking the data from intersections of mean lines in the manner already described. The results from the spading done by no two men will agree, and the same man cannot often check himself. But nevertheless the author feels that the proof of the laws which exist between the quantities involved will be of the utmost assistance when supplemented by such actual field data as can be made universally applicable to the needs of engineers.

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THE DISCUSSION

E. F. NORELIUS:—Investigations of this kind will be of considerable value in enabling tractor engineers to compare results obtained with different methods of drive, without actually making extensive field tests, although it would be necessary to make some preliminary tests.

With a proper foundation, however, it is possible to develop the relations between different functions. In this connection I have had in mind a different manner of analyzing the forces involved. Referring to Fig. 1 of Mr. Moyer's paper and using the same letters, I have assumed that the ground pressure under the wheel must be normal to the surface of the wheel at all times, and that the bearing value (n) of the soil is proportionate to the depression of the soil. To simplify the problem and readily analyze the forces, the half segment bounded by the lines a and d and the circular portion of the wheel may be considered as filled with a liquid, whose density per cubic inch equals n. This liquid will produce pressures on the inside of the rim exactly equivalent to the exterior pressures, and therefore will bear all forces which previously were in the rim when it was in contact with the ground. The value of P will then be equal to the horizontal components of the pressure of the liquid on the rim, which is equal to the pressure of the liquid concentrated on the element d. The unit pressure at the lowest point of the wheel equals dn, therefore,

$$P = \tfrac{1}{2}\, d^2\, Fn \qquad \text{(a)}$$

The weight supported by the wheel rim is equal to the weight of the liquid which we have put into the rim, or,

$$W = AFn \qquad \text{(b)}$$

where A represents the area of the half segment of the circle below the ground.

In the equations (a) and (b), we have a value of P and of W. The value of P agrees with equation (6) of Mr. Moyer's paper, which, so far as the wheel is concerned, is dependent upon the depth of depression and the face, and which we can readily see is correct, as the work done with a certain depth of depression of the wheel, with certain ground conditions, is dependent entirely upon how fast we travel ahead. From the above, we can determine a value of G or equivalent grade:

$$G = \frac{P}{W} = \frac{d^2}{2A} \qquad \text{(c)}$$

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I have not had the opportunity to check the above value of W and of G with the equation derived by Mr. Moyer. I believe that my analogy in connection with the use of the liquid is correct and gives a very simple method of analyzing the various functions.

A. W. SCARRATT (M. S. A. E.):—The investigation carried on by Professor Moyer is, I believe, along correct lines, and when embodied in simple formulas should prove very useful. These investigations should by all means be continued, as the really valuable information is still to be obtained. Future experiments with tractors of known mechanical efficiency should be conducted in the field with a view to determining the coefficients of traction of smooth wheels on various soils and also, in each case, the limit of tractive effort under given soil conditions for wheels equipped with various types of cleats and grouters, when working under loads. The increased resistance to rolling of wheels equipped with cleats or grouters of various forms should also be determined.

These experiments should be conducted in the open, under the guidance of a committee of engineers, by men paid to devote their time to this work. Even when done in this way considerable time will be required to obtain all of the data. The data would be very valuable to the various universities, and their financial assistance should be solicited.

L. F. SEATON (M. S. A. E.):—Tractors being a comparatively new development in the automotive industry, little is as yet known of their actual performance under varying conditions. The work conducted by Professor Moyer is another step in the advancement of this new engineering phase. Such work should be encouraged not only because of its scientific importance, but also because of its significance in the future development of the tractor.

Professor Moyer has shown conclusively that the resistance between wheels and the soil increases with velocity, the rate of increase being somewhat more rapid with heavy loads than with light loads. Lines plotted to represent the relation of speed to the resistances, for weighted wheels under varying conditions were shown to intersect at a point of zero resistance and a speed (expressed in revolutions per minute) of a definite negative quantity for all wheels. Of course it is difficult to comprehend the full meaning of such re-

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sults. Whether they actually represent practical data or merely hypothetical results that may be verified mathematically is as yet to be determined. That the bearing value of the soil is dependent upon the speed and diameter of a wheel was another interesting result verified by these investigations.

GEARS FOR TRACTORS

By A W SCARRATT*

The problem of gear teeth is one of the biggest that we have in machine design. To be sure it has had no small amount of investigation, but it is still in a chaotic state. Real data of an engineering nature, which are still lacking, would be of great value to all concerned with gearing problems and would aid greatly in conserving valuable material.

The subject will be treated mainly as it applies to tractor engineering problems, but first let us briefly consider the modern automobile and its transmission. About 95 per cent of the time that the automobile is in use, the power is delivered directly to the differential gear, the transmission gears not being called upon to transmit power. During the remaining 5 per cent of the time the transmission is required to deliver this power. Therefore, the actual period of use to which the transmission is put is exceedingly small, when compared with the actual life of the car. Even during this time it is very seldom required to transmit the full power of the engine. It is because of these conditions that automobile transmissions can be built so lightly.

AUTOMOBILE VERSUS TRACTOR TRANSMISSIONS

The materials and workmanship, no doubt, are excellent, but the factor of safety is low when figured for full-load torque, and the factor of wear is still lower. The contrast is certainly great between these conditions and the demands put upon tractor transmissions and gears.

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A tractor will be used for approximately 1000 to 1500 hours in a year's time, which is about three times as long as the automobile is used, if driven 8000 miles in a year's time, and the transmission is required to deliver the power during the entire period of operation. It is safe to assume that during this time the tractor will operate continuously at not less than 80 per cent of its full-load rating. The automobile generally operates at from 15 to 25 per cent of its rated capacity. A tractor transmission is required to do about 200 times as much in a year as the average automobile transmission, or the tractor transmission sees more service in one year than the average automobile transmission would in 200 years, hence, when designing tractor transmissions, the most careful consideration and study are necessary. It is true that tractor transmissions have been heavy and crude, and that as a result the weight of the machines is high and the efficiency low, but it must be borne in mind that the large gasoline tractor of today is an outgrowth or revision of the steam threshing and road engine of twenty years ago, plus improvements, and not an offspring of the automobile. However, one cannot but realize the advantages of refined engineering and quality of materials and construction, and also of standardization, as exemplified in the modern automobile, and it is for these reasons that there is such a revolution in tractor design.

Materials for Gears

From the foregoing it will be seen, then, that when laying out the gearing, careful judgment and good engineering are essential, if high efficiency, minimum weight, and satisfactory service are to be obtained. Speed, torque, and space limitations, manufacturing facilities, and selling price, all have their influence on the type and the material of gears.

Of material the engineer has a considerable choice, and he also has several types of gear teeth from which to choose. Materials for gears range from paper, wood, cloth, hide and compositions to the finest alloy steels. Each has some advantage, be it strength, cost, operation, ease of production or wearing quality, and each tooth shape has some point in its favor.

Table I shows the tensile strength, elastic limit, machining properties, and other characteristics of several common materials suitable for tractor gears. The varia-

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TABLE I—MATERIALS FOR GEARS

Material	Tensile Strength, Lb. per Sq. In.	Elastic Limit, Lb. per Sq. In.	Machining Quality	Characteristics
Close-grain cast iron	24,000	.....	Cuts freely. Good surface	Easily obtained—low cost—satisfactory for low pressures.
Close-grain, semi-steel	31,000	.....	Cuts freely. Good surface	Easily obtained—low cost—satisfactory for high or low speeds and low pressures—wears well.
Cast steel	60,000	31,000	Machines well when annealed	Medium cost—strong and tough—loss due to defects sometimes large. Wears well. Satisfactory for low speeds and low pressures.
*0.20 carbon steel (S. A. E. No. 1020)	75,000	43,000	Care needed to avoid tearing	Affords good combination of strength, toughness and hardness—good for high speeds and medium pressures.
†0.35 carbon steel (S. A. E. No. 1035)	95,000	65,000	Machines well when annealed	Good combination—strength, toughness and hardness. Good where high speeds or heavy pressures prevail.
†0.45 carbon steel (S. A. E. No. 1045)	110,000	75,000	Machines well when annealed	Excellent strength but more brittle than the previous steel. Suitable for high speeds and for steady heavy pressures. Wears well.
†3½ per cent nickel steel (0.20 carbon)	130,000	99,000	Hard to cut. Fine surface	Excellent strength, toughness and hardness. Very satisfactory for high pressures.
†3½ per cent nickel steel (0.40 carbon)	155,000	130,000	Hard to cut. Fine surface	Excellent strength, toughness and hardness. Suitable for very severe conditions. Good for clash gears.
†Nichrome forging steel	150,000	130,000	Machines well	Same physical advantages as above—easily heat-treated. Excellent wearing qualities.
Chrome vanadium steel	190,000	150,000	Machines well	Same as previous steel.

*Carburized and heat-treated. †Heat-treated

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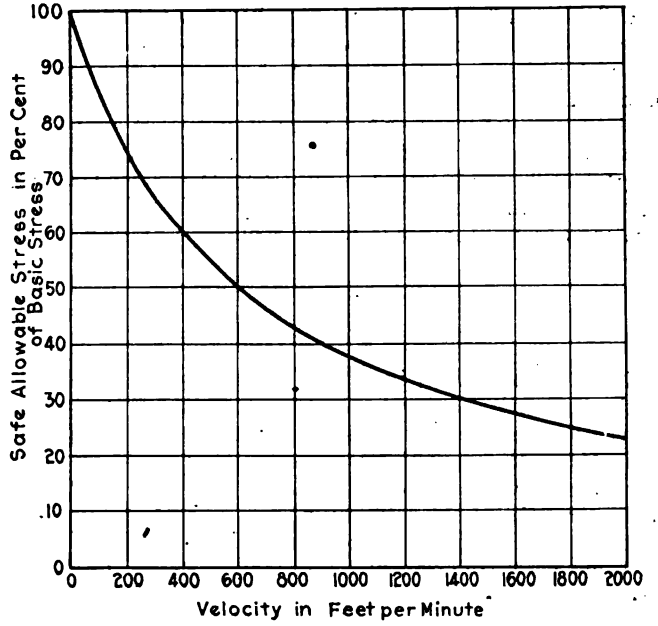
tion in strength is great, but it is even greater in practice than these figures show; when using metals of low strength, the factor of safety must be larger than is required when using metals of greater strength, because of the variance in physical properties to be found in the lower grade materials. The physical properties of any of the carbon steels and alloy steels can be varied considerably to suit the designer's requirements, so that the fig-



VALUES FOR FACTOR S IN THE LEWIS FORMULA FOR STRENGTH OF GEAR TEETH

Safe working (basic) stress for various materials, lb. per sq. in.			
Rawhide	5,000	Bronze	12,000
Cast iron } Cloth }	8,000	Cast steel	20,000
Semi-steel	10,500	Forged steel (not treated)	25,000

ures given do not apply to all cases. The strength desired is regulated in the heat treatment by the drawing temperature used, the low drawing temperatures giving steels of high tensile strength, high elastic limit, and good hardness, but at a sacrifice of some of the toughness. The higher drawing temperatures decrease the tensile strength, elastic limit and hardness, but will materially

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increase the toughness. Therefore, in choosing the material for a gear, careful consideration must be given to securing the proper strength for the loads imposed upon it, proper hardness to insure reasonable service and proper dynamic properties to resist impact and fatigue. Here is where we lack much information.

Certain materials seem to be satisfactory for automobile gears, in view of the small amount of service they get, but it is questionable if gears made of similar proportions and of the same materials working under full-load torque all the time, and, therefore, under much greater surface pressures, will last as long as they should in tractor service. It seems to the author that experiments should be made to determine definite data on the relation of surface pressures and the degree of surface hardness necessary for a given length of service. These properties must bear some reasonable relation to one another for any given material. Certainly private investigations have been made which would be of great value if given to the engineering profession.

Strength of Gear Teeth

There is also much information lacking on the subject of the strength of gear teeth. On account of the various shapes of teeth, no common rule can be used in figuring their strength and the problem becomes a complex one. For instance, we have the Brown & Sharpe 14½-deg. tooth, the Grant 15-deg. tooth, and a 20-deg. tooth, all of which are of equal length and agree in all proportions excepting the pressure angle. This, of course, varies the strength of the tooth in the ratio of the squares of the root thickness. The root section of the tooth is the critical one when figured as a cantilever beam, the load being applied at the outer edge of the tooth. In a rack gear of any given width, the 20-deg. pressure angle tooth is practically 25 per cent stronger than the 14½-deg. tooth. This ratio is not constant, however, in fact, it becomes slightly greater as the number of teeth in the gear becomes less. This is one advantage in favor of the 20-deg. pressure angle.

The most commonly used formula for these particular types of teeth is that originated by Wilfred Lewis, where

$$W = SPFY \frac{600}{600 + V}$$ in which

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W = Safe tooth load in lb.
S = Safe stress in lb. per sq. in.
P = Circular pitch, inches
F = Face, inches
Y = Factor depending on the number of teeth in the gear and the pressure angle used.
V = Velocity in ft. per min.

The factor Y is governed entirely by the number and shape of the teeth. Tables of these values are given in all engineering handbooks. The values of S for different materials and velocities are given in the accompanying diagram. These values I believe are subject to criticism because the allowable stress drops with an increase in speed in the same ratio for every material regardless of its physical properties. This certainly should not be the case. When we consider the ability of materials, such as chrome vanadium steel, to resist impact forces, and the inability of other materials, such as cast iron, their classification in the diagram seems foolish. In fact, several large manufacturers do not adhere to the Lewis recommendations for the allowable stress.

Previously, when considering the strength of the gear tooth, attention was called to the value of increasing the pressure angle. Another way of increasing the load-carrying capacity of the tooth is by shortening it in a manner similar to the Fellows so-called 20-deg. stub tooth standard. The designers of that system realized the value of the short tooth and made a system of tooth sizes in which the addendum, dedendum, and clearance of a small tooth are combined with a tooth of large pitch. The Fellows stub tooth system comprises the following tooth sizes: 4/5, 5/7, 6/8, 7/9, 8/10, 9/11, 10/12 and 12/14.

It will be seen that there is no definite ratio between the height of the tooth and its pitch, such as prevails in the Brown & Sharpe system.

Another disadvantage with the Fellows system is that the range of tooth sizes is not adequate, especially in tractor work. The author has been forced on several occasions to use stub teeth of larger pitch than any given in the table, these sizes being the ¾ and 2/2½ pitch. Another disadvantage in the present stub tooth system is that no convenient or accurate formula is available for figuring the strength of the teeth, and there cannot be one until considerable investigation has been made. Why should we not have a standard 20-deg. stub tooth which has definitely fixed proportions based on the pitch, such

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as is exemplified in the 20 deg. stub tooth proposed* by Charles H. Logue.

The customary practice has been to figure the strength of the regular 20-deg. tooth and then increase this strength in the proportion of the inverted pitch fraction of the stub tooth. This does not give the correct strength, as it assumes the root thickness of the full-length tooth and stub tooth to be alike, and this is not the case. The root thickness of the 20-deg. stub tooth is less than that of the 20-deg. full-length tooth of equal pitch measure, and consequently it has a smaller section modulus.

THE DISCUSSION

H. C. BUFFINGTON (M. S. A. E.):—It seems to me that there is just as much difference between cast iron and chrome vanadium steel as between steel and wood; it is my opinion that in the standardization of gear teeth engineers always have fallen down. There is no standard. When we instruct a draftsman to lay out a gear he probably will take some standard that we did not have in mind at all; and when we tell him to lay out a stub-tooth gear and have another man check up his drawing, they will each have a different result. Another person checking the same drawing will probably get another result, so it seems that teeth forms are still unstandardized.

I do not see why engineering societies cannot jointly adopt some form of stub tooth in order that we may have interchangeable gears. The electric starter on an automobile is small; it is really too small for the larger engines, if it is going to be used on tractors in the future. Not long ago I was laying out the gear teeth for an electric starter on a large engine and I found that it was necessary to employ a 7/9 pitch or one a little larger. That is drawing right away from the automobile form because in order to get a tooth that is strong enough for cast iron it is necessary to use the larger stub tooth, and then the question arises as to what stub tooth we shall use. I suggest that we outline some standard tooth before going too far into the electric-starter proposition, and the same remark applies to many other gear forms and transmissions.

CHAIRMAN E. R. GREER:—If standardization is necessary and to be adopted there must be teeth of the same

*In *American Machinist*, June, 1907

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pitch on every tractor flywheel if we are to make starters interchangeable. Starter manufacturers have standardized small steel ring gear teeth for automobiles.

Once done, naturally the starter makers would want to use the same standard for the larger pitch, but the larger pitch would not be a standard that any automobile manufacturers are now using.

Even in transmissions, where each tractor company is making all its own gears, there would be quite an advantage in a standard tooth. In purchasing cutters and in checking teeth there would be a great advantage in having all gears the same.

G. C. ANDREWS (M. S. A. E.):—In reference to the chilled-iron tooth, we get wearing surface on our coarse work and strength and hardness, but we do not get toughness. I would like to know if there have been any experiments along these lines.

When we reduce pressure surface, will it not become so small that the lubricant will have little life, soon becoming spoiled? Last summer I was looking up some tests for transmissions and it was a surprise to learn that there is almost no gain in lubrication when using a large gear.

F. MCDONOUGH:—When we heat-treat gears we find that the loss is much less with the stub-tooth gear than with the old long tooth. There is a great tendency to use stub teeth for this reason.

A. KRIEG (M. S. A. E.):—Some time ago I asked tractor companies about the addendum and dedendum of their teeth and they told me these were three-quarters of a standard tooth; the clearance is the same as the clearance of the standard tooth, which is one-twentieth of the circular pitch. I do not see why having the addendum and dedendum three-quarters of a standard tooth would not be satisfactory for tractor work. It is generally used now. I believe all cutters are standard, and if we change now to another standard it would be difficult to get cutters.

CHAIRMAN GREER:—I think you are right about the impracticability of any change; the matter has gone so far that the Fellows' standard is all we can use. Are you sure about the proportions?

MR. KRIEG:—They are not exactly right. With the larger ¾ pitch, three-quarters of the ordinary tooth can be used. Take the double addendum, multiply it by 0.75 and then make the clearance the same as on a standard tooth, which is one-twentieth of a circular pitch. I think that works out well.

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MR. BUFFINGTON:—Would you express the stub tooth pitch as a fraction? Would you not have to express it as a decimal?

MR. KRIEG:—That is one mistake they have evidently made.

CHAIRMAN GREER:—It works out pretty well until we get into the half pitches in the large sizes. It is a big jump from the 3 pitch to the 2 pitch. Naturally 2½ pitch must be used but when we get into half pitches then we strike a peculiar combination.

MR. SCARRATT:—If we have a massive gear that we want to heat-treat to get a good surface hardness, we shall in all probability have to change the drawing temperature and also the medium used in drawing. Oil, brine and plain water are used. The oil does not have the quick effect on the treatment of the gear of either brine or water, and in most of the carbon steels listed the drawing temperature would range probably anywhere from 450 up to 1200 deg. and the drawing temperatures used in these stated values would be about an average of the two. If the maximum strength and hardness are desired in a gear and the dynamic properties are not to be given such great consideration, for instance where the loads are very constant, a lower drawing temperature could be used, which would give high tensile strength and elastic limit and a harder gear. But in tractor service we have a severe problem to contend with. The shocks are great and they are intermittent, and as the tractor might be operated for long periods without rest it is not safe to work gears up to the maximum tensile strength and elastic limit. We must consider seriously the toughness of the steel and its ability to withstand the impact loads that we get, and that will necessitate drawing the steels at a little higher temperature than for the other stated condition. The exact range needed for those gears would be hard to determine. The values given in the paper for the carbon steels and the nickel steels were taken from the specifications in the S. A. E. Handbook and are about medium values throughout.

Factors of Safety

A MEMBER:—What factor of safety is used?

MR. SCARRATT:—That is a matter of opinion. If the gears are put into very severe service where the shock load is going to be tremendous, such as in tractor work, it is not unreasonable to suppose that at times the load on a tractor will exceed the maximum rated load of the engine by 100 per cent. If that were true, in other

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words, if we did not think that the shock load would set up such a high stress in the gears that the brittle point would be reached, but that there would be a permanent set and a possibility of rupture in the teeth of the gear, then a basic stress at least 50 per cent lower than the elastic limit given would have to be used. If gears are put into use where they are not likely to meet such conditions as that, stresses which run nearer to the elastic limit can be used. That is largely a matter of judgment. Definite figures cannot be given. In view of all this, treating with the lowest safe working stresses is conservative.

MR. ANDREWS:—Which is conservative, the 4, 5 or 6?

MR. SCARRATT:—In tractor work I should say that if stresses were taken as, say, 50 per cent of the elastic limits of the stated material, they would not be too high for the different gears. That stress would be for a static load—practically no speed—and if stresses in accordance with the values plotted in the Lewis curve were taken we would be very safe at the higher speeds owing to the ability of these better grades of steel to withstand shock as compared with such materials as cast iron and semi-steel for which that curve is suitable.

MR. ANDREWS:—If a spring is put in to take care of shocks, as on the new Foote tractor, can any allowance be made?

MR. SCARRATT:—I believe not.

Effect of Wear Factor

CHAIRMAN GREER:—When values for wear are figured out, is it not true in almost every case that the teeth are considerably larger than needed for tensile strength. Does not the wear factor insure having sufficiently large teeth?

MR. SCARRATT:—We know less about the factor of wear in gears than we do about the strength of materials.

CHAIRMAN GREER:—If we design a gear for a known load and figure the factor of safety as 2, that would be 50 per cent of the elastic limit and I am sure we would have a rather small tooth for anything we might want to consider in a tractor. We would probably decide to have it four or five times as large as that unless we were using cast iron. With good steel it would be small.

A. F. MOYER (M. S. A. E.):—How would one start out to experiment as to wearing of gears and gear teeth?

MR. SCARRATT:—As stated in the paper, I think we know less about the factor of wear than of any other gear factor. Automobile transmissions have stood up

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well. The gears will probably last longer than the machine, as a rule, and their life is long because the work done is relatively so little in comparison with the life of the machine. If we put gears of similar proportions to those used in automobile transmissions in tractor transmissions which, as I stated, would get possibly 200 times as much service in a year as an automobile transmission, it is questionable if the gears would last anything like as long. Gear and transmission manufacturers certainly must have run gears of given sizes and of various materials under maximum loads for a long period of time in order to determine the amount of wear and the surface pressures that they could reasonably stand, but no information of that kind is available.

It would not be a hard problem to figure the strength of a gear and compute the necessary size if we only knew how much pressure that gear was able to take under continual service and still stand up for a reasonable length of time. The only way that I know of for finding it is to take sets of two small gears of various sizes and of various materials, first test the materials, and then run them continuously for a thousand hours under the rated load for which they are computed. We could then measure the amount of wear on the teeth.

Testing Gears

MR. MOYER:—I have looked up all the data I could find from various sources. So far as I know, the published experimental data are based entirely on the running of cast iron gears. A proposal was made to test gears by running them in some machine that would place pressure on the teeth, but this could not solve the entire problem. If we take four gears we can mesh all four so that they will fit in one position only, depending on the number of teeth. To distort the centers, we move any of the gears a little and apply pressure. If we were to take a chain of gears like that and apply pressure diagonally on opposite sides of the square and run them for a long time to find the amount of power necessary to overcome the friction, I think we could get some good data on the pressure.

MR SCARRATT:—We put a set of bevel gears in the differential of a machine and gave that machine a long run. It had approximated 350 hours of continuous service, all told, when we removed the differential from the machine to see the condition of the gears. They were stub tooth gears of the best semi-steel we could make and were very accurately cut. We found that

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in 350 hours of service a full thirty-second of an inch of metal had been removed from the faces of the teeth. They were worn perfectly as to shape. The action was just a crushing from the gear surface. Now, with any amount of figuring, that would appear to be a perfectly safe gear and it was running submerged in oil continuously, but the surface pressure was in excess of what the actual strength of the material would stand.

MR. MOYER:—What were the pitch and tooth pressure of those gears?

MR. SCARRATT:—It was a 4/5 stub tooth bevel gear, 12-in. pitch diameter. There were four pinions in the differential and I believe that the torque on the mean pitch radius would be approximately 2800 or 2900 pounds. The face of the gears was 1¾ in.

Vibration and Dirt

CHAIRMAN GREER:—In tractor work the vibration and dirt conditions must be considered. In any laboratory test gears would be run clean and it would be foolish to make a test that way. Running them without vibration would make a tremendous difference. Slight vibration might make a difference of three or four times in the life of the gear. As Mr. Scarratt stated, the gears that he mentioned failed from crushing and if a gear will fail from crushing, vibration would certainly affect the wearing of the material.

MR. SCARRATT:—In modern work we have arrived at the conclusion that cast iron has no place in any kind of transmission or gears, that is, for tractors designed for the minimum of weight and the maximum of power and efficiency.

A MEMBER:—What about the use of manganese in gear material?

Ferro-Manganese

MR. HATFIELD:—Ferro-manganese contains about 15 per cent of manganese. It is a manganese and iron alloy of great strength and is being used quite generally in all sorts of mechanical constructions where resistance to wear is required. It undoubtedly would be an excellent material for gear teeth where resistance to wear is required.

A MEMBER:—What about machining?

MR. HATFIELD:—It is absolutely non-machinable.

A MEMBER:—That eliminates it for cut gears.

MR. HATFIELD:—It has to be cast. It has a ten-

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sile strength of about 80,000 lb., an elastic limit of about half that, and is comparatively soft (hardness about 250) but is wonderfully resistant to abrasion.

Cast Iron Inserts

O. W. YOUNG (M. S. A. E.):—In tracklaying types of tractors, particularly in the case of the Yuba and Best, manganese drive sprockets are used. Some holes must be drilled in the hub bearings and inserts of cast iron are cast in the manganese steel at those places, and the holes are drilled in the cast iron. Whenever anything must be fastened to the gear a little cast iron is employed in this way.

CHAIRMAN GREER:—Is the track with which this gear engages of manganese steel?

MR. YOUNG:—No. Different materials have been used. Manganese track rings have been used for dredges, but they were not necessary. An alloy steel bull pinion is used.

CHAIRMAN GREER:—Do they machine the pinion and cast the gear?

MR. YOUNG:—Yes; the pinions probably wear out five to one.

J. N. NEAD (M. S. A. E.):—Referring to Mr. Moyer's suggestion of a mechanism for testing gears I recall the time a test on gears was under way at an arsenal. They were having much difficulty with traversing gears on disappearing carriages. They had a gear train rigged up, consisting of four gears arranged in a rectangle. Gear 1 was driven by a belt, gear 1 drove gear 2, gear 2 drove gear 3 through a shaft, and gear 3 drove gear 4. There was a drum between gear 1 and gear 4 with a spring in it which applied tension to the gears throughout the train. It struck me as a good method of testing gears. It was mounted in a basement close to a coal bin where there was plenty of dirt.

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PRINCIPLES OF TRACTOR ENGINE COOLING

By Arthur B Modine*

There are many factors other than radiator size involved in tractor engine cooling. The principal variables pertain to radiator type, radiator core, thickness or degree of cooling capacity, rate of water circulation, size, type and speed of fans, and the economy characteristics of the engine to be cooled. I do not consider it practical to evolve a formula covering all the branches of gas engine operation, but as I believe the time opportune for a certain degree of standardization of the variables referred to, I shall go into some of the essentials of cooling involved in our particular branch of the automotive industry, and, in conclusion, will give a formula for radiator sizes and recommendations for water circulation, air velocity and fan details.

Gas engine cooling involves a diversity of elements in engine design, all affecting the economy of the engine, and ultimately, for our purpose, that proportion of heat input showing up as jacket loss.

Equation I is our introductory and basic formula, in which Heat to radiator = Heat input — Heat equivalent of horsepower developed — Exhaust loss — Engine radiation and convection losses. (1)

All the factors of this equation for any given engine can be computed, but for general use certain assumptions are required together with the introduction of a variable factor.

Equation II is based on observed average performances.

$$\text{Heat to radiator} = 0.40 \text{ heat input} \times F \qquad (2)$$

where F is a variable with a value of 1.0 in the case of L-head engines ranging from 25 to 50 hp. and 0.8 in the case of valve-in-head engines of the same power range. Individual peculiarities of jacket design, piston speed, compression, proportion of engine radiation and convection loss, and valve design, would all, of course, further affect the value of the variable, but I believe that the formula, as suggested, will be found sufficiently accurate for our purpose.

*President of the Modine Mfg. Company

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Taking average tractor engine economy as 0.75 lb. of gasoline per hp. per hr., and a heat value per pound of gasoline as 20,000 B.t.u., we find that Equation 2, applied to the L-head type engine, becomes

Heat to radiator per min. per hp. =

$$\frac{0.40 \times 0.75 \times 20{,}000 \times 1}{60} = 100 \text{ B.t.u.} \qquad (3)$$

Hence, if we are to cool a 30-hp. L-head engine under full load, it is necessary to provide cooling capacity equivalent to $30 \times 100 = 3000$ B.t.u. per min.

It is interesting to note how this quantity of heat would manifest itself were no means available to dissipate it other than the evaporation of water. The 3000 B.t.u. per min. would evaporate about 3 lb. of water per min., 180 lb. or about 21½ gal. per hour. Double this to 43 gal. per hr. for, say, a 60-hp. load, and we get an idea of the difficulties that confronted the pioneers in the tractor industry in getting sufficient tank capacity. Aside from other considerations, it is manifestly impracticable to carry around a great bulk of water such as would be required to absorb this amount of heat through evaporation, and to meet these conditions it has been found necessary to transfer this waste heat to the surrounding atmosphere through the medium of the conventional radiator and cooling system shown in Fig. 1.

This comprises a radiator, generally a circulating pump, and a fan, the radiator consisting, for the purpose of our discussion, of a multiplicity of tubes conveying the water to be cooled and spaces around these tubes for the passage of air. Referring to Fig. 1 let us consider the factors involved in the transfer of heat from the engine, through the radiator and into the atmosphere. The water from the engine is delivered to the radiator at a temperature T_1 and must be cooled to a lower temperature T_2. The heat then taken from the engine or from the water will be in terms of the weight of water circulated (W_w) times the temperature drop ($T_1 - T_2$) effected upon it by its passage through the radiator and may be expressed as $W_w(T_1 - T_2)$. It is evident that the heat removed from the water must (under normal operating conditions) equal the heat imparted to the air, and will be measured in terms of the weight of air circulated (W_a) times the temperature rise ($T_4 - T_3$) effected upon it by its passage through the radiator, or $W_a(T_4 - T_3)$.

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From this balance we get Equation 4,

$$W_w(T_1 - T_2) = W_a(T_4 - T_3) \quad (4)$$

which, expressed in foot-pound-second units, becomes in equation V

$$W_w(T_1 - T_2) = V_a \times 0.071 \times 0.2375(T_4 - T_3)$$
$$= 0.01685V_a(T_4 - T_3) \quad (5)$$

in which W_w is weight of water in lb.; V_a is volume of air in cu. ft.; 0.071 is the weight of 1 cu. ft. of air at 100 deg. fahr.; 0.2375 is the specific heat value of air.

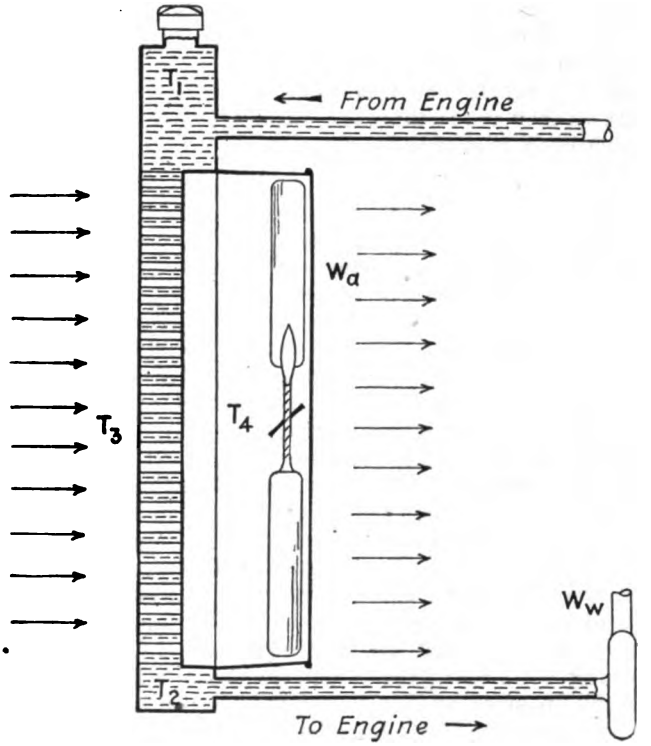

FIG. 1—CONVENTIONAL RADIATOR AND COOLING SYSTEM

Equating this last relation with Equation 3 we get

Heat to radiator per min. per hp. = 100 B.t.u.

$$= W_w(T_1 - T_2) = 0.01685V_a(T_4 - T_3) \quad (6)$$

from which we derive

$$W_w \text{ (or lb. water) per min. per hp.} = \frac{100}{T_1 - T_2} \quad (7)$$

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V_a (or cu. ft. air) per min. per hp. =

$$\frac{100}{0.01685(T_4 - T_3)} = \frac{6000}{T_4 - T_3} \qquad (8)$$

and from Equation 8, inasmuch as the volume of air is a function of its velocity and area, we get

$$\left.\begin{array}{r}\text{Square feet frontal area} \\ \text{of radiator per hp.}\end{array}\right\} = \frac{6000}{v(T_4 - T_3)} \qquad (9)$$

in which v is the air velocity in feet per minute. These values are all dependent upon the value of the factor F in equation 2. This factor has been omitted to avoid the use of too many symbols, but in view of its value of 1 for L-head types, the formulas will apply directly as they are to this type, while for the valve-in-head classification they are to be multiplied by the factor F with value 0.8, as designated.

From an analysis of Equation 9 we see that our problem of cooling or of radiator size has resolved itself into two factors: the temperature rise effected upon the air and the velocity of the air. The first item is most important to our discussion, but involves such a mass of detail that I will deal with it briefly only in its relation (1) to water delivery and (2) to air velocity. We have a physical law that the amount of heat given off from a source is proportional to the difference in temperature between the radiating body and the air, and other authority states that the loss of heat by convection is nearly proportional to the difference in temperature between the hot body and the air. This being true, we are concerned in maintaining the surfaces in a radiator at an average temperature as high as is practicable, and this, of course, in our case is directly a function of the rate of water circulation. It becomes a matter then of choosing a standard for circulation that will give practical pump sizes and still maintain a reasonably small temperature drop through the radiator. With this in view, I propose as a standard that rate of circulation which will give a temperature drop to the water circulated through the radiator of not more than 15 degrees fahrenheit. From Equation 7, then, we get

$$\text{Pounds water circulated per min. per hp.} = \frac{100}{15} = 6.67 \qquad (10)$$

$$\text{Gallons water circulated per min. per hp.} = 0.8 \qquad (11)$$

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This would give a circulation per minute of 16 gal. for a 20-hp. engine, 24 gal. for 30 hp., etc.

Effect of Circulation

Fig. 2 will serve to illustrate the effect rate of circulation has on cooling capacity. In the first radiator water enters at a temperature of 200 deg. and is circulated at the rate of 200 lb. (about 24 gal.) a minute, leaving the radiator at 180 deg. The heat given up amounts to 4000 B.t.u. per min., and the average temperature of the radiator surfaces is 190 deg.

In the second case conditions are the same except the circulation, which is only 100 lb. per min., requiring a 40 deg. drop to get rid of the same amount of heat as in the first case. The fan is not drawing any more air; consequently, in order to absorb the same amount of heat as in the first case, the average temperature of the surfaces must be maintained at 190 deg., which requires an entering water temperature of 210 deg. and a leaving temperature of 170 deg.—or all the difference, with the same radiator, between satisfactory cooling on the one hand and boiling on the other.

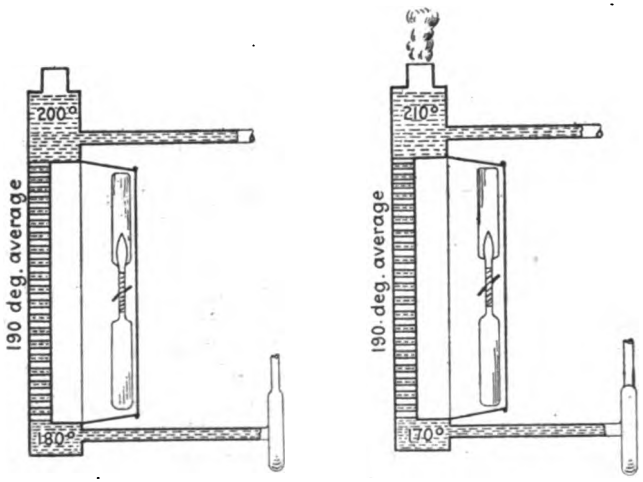

FIG. 2—EFFECT OF RATE OF CIRCULATION ON COOLING CAPACITY

The relation of temperature rise of the air to velocity of the air over the heated surfaces will have to be left with the brief statement that, as we get up into the

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velocities practicable for our purpose, the heat taken up by convection increases almost directly in proportion to the velocity. This is illustrated by the temperature rise curve, Fig. 3, which is typical of the temperature rise characteristics of almost any type of radiator, as well as the thickness of radiator core. The only limit that we are bound by in air velocities is the power available to effect the velocity, but with this in view, as well as further remarks to follow, I would propose a standardized air velocity referred to gross radiator core area of 2000 ft. per min. This velocity is practicable with most of the radiator types on the market and, as applied, will meet nearly all requirements up to 50 hp. in space available for radiator mounting. The matter of a definite value for the factor $(T_4 - T_3)$ concerns the characteristics of design of each type of radiator core and its thickness, but most of them intelligently offered for tractor work will be able to meet the requirements of a 30 deg. rise with initial or operating temperature of 100 deg., air velocity of 2000 ft. per min., maximum water temperature of 200 deg. and water circulation as suggested. I say "intelligently" as regards experience with the limitations of space available for radiator mounting. These assumptions meeting approval, Equation 9 comes to

Sq. ft. radiator required per hp. =

$$\frac{6000}{30 \times 2000} \times 1 = 0.1 \text{ for L-head engines.}$$

Sq. ft. radiator required per hp. =

$$\frac{6000}{30 \times 2000} \times 0.8 = 0.08 \text{ for valve-in-head engines.}$$

From some fifty cases I have tabulated data as to rated horsepower (using the formula $\frac{D^2SN}{F} \times$ No. cyls., in which F has a value of 12,000 for valve-in-head and 13,000 for L-head types), actual radiator size, fan size and fan speed in each instance. These cases I have divided into groups of varying horsepowers and averaged the data in each group. In applying the suggested formula for radiator size, I find that it would check well in the 20-hp. group if in a few of the cases the 18-in. fans were run at a higher speed than 1450 r.p.m. In the 30-hp. division L-head type, the actual radiator size is about 12 per cent larger than that proposed by the formula, but the average fan of 19 in. diameter was driven at only an average speed of 1600 r.p.m. In the 30-hp. valve-in-head class the formula radiator size is about 3 per

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cent larger than actual sizes with average fan size 18.66 in. and average fan speed 1866 r.p.m.

In the 35-hp. L-head class the formula radiator-size is about 3 per cent larger than the actual size. Average fan size is 19.3 in. and average fan speed 1666 r.p.m. Some of the cases in this division, I know, are not cool-

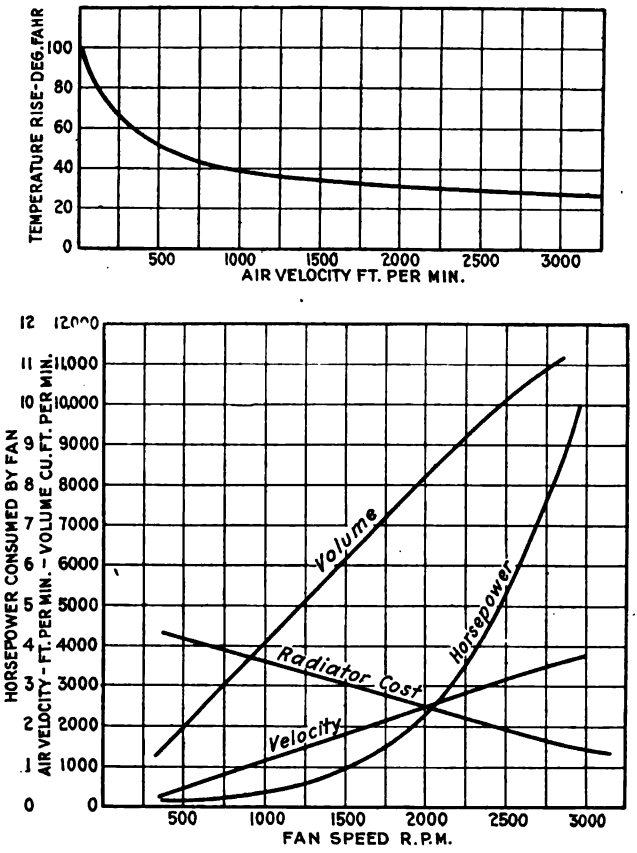

FIG. 3—UPPER DIAGRAM SHOWS RELATION OF TEMPERATURE RISE OF AIR TO VELOCITY OF THE AIR. LOWER DIAGRAM SHOWS CHARACTERISTIC CURVES OF A TYPICAL 20-INCH FAN AT VARYING R.P.M.

ing properly, and the division as a whole could well afford to use larger fan diameters. In the 40-hp. class the formula radiator-size checked with the actual size. One 50-hp. case, a well-known tractor, gives formula radiator-size about 13 per cent larger than actual size. This

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tractor is considerably overpowered, but its cooling performance under average conditions is entirely satisfactory. I know, and its makers know, however, that at heavy full load boiling will occur, and they agree with me that the 13 per cent additional, as recommended, is justified.

One 64-hp. valve-in-head engine tested a short time ago has actual radiator size about 4 per cent greater than formula size, but the cooling capacity was found to be at least that amount in excess of requirements. As we get up into higher horsepowers, the space available in which to mount the radiator becomes a more and more important factor, so that air velocities and fan speeds must be increased at the expense of added power for fan drive. This introduces an added variable in our formula and consequently the radiator size formula is not recommended for use above 50 hp. The basic equations may, however, be adapted to larger outputs.

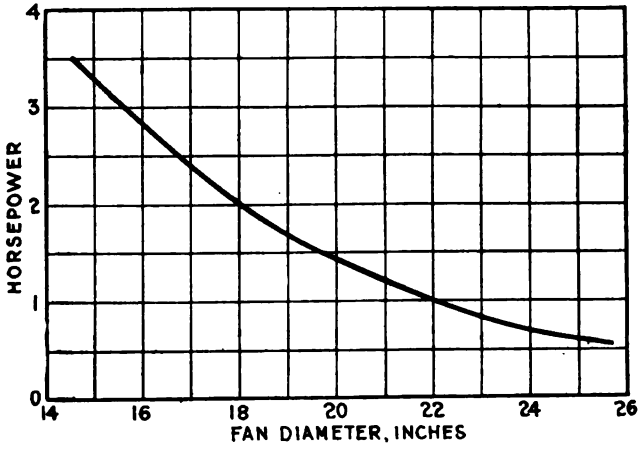

FIG. 4—HORSEPOWER TO DRAW AIR AT 7000 CU. FT. PER MIN. THROUGH RADIATOR

By the foregoing it will be seen that the problem of cooling is fundamentally one of how much air can be economically delivered through the radiator, and to what degree it can be economically heated. Granted efficient fans to choose from, it would seem that the first consideration should be the amount of power that we would be justified in using to effect the delivery. This available power could conveniently be expressed in terms of engine horsepower.

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Fig. 3 shows air deliveries effected by a typical 20-in. fan at varying fan speeds, corresponding air velocities and power consumptions, the characteristics of each of which are typical of fans such as are offered to the trade. It will be noted that the volume of air that can be handled and its corresponding velocity increase well in proportion as the speed increases, but it is evident from the power consumption curve that a certain point is reached in fan speed beyond which the power consumption mounts upward with astounding rapidity.

If it were not for this, we could deliver so much air through the radiator that there would not be much left for the radiator manufacturer to do, as the radiator cost curve indicates, but in this event, of course, the purpose of the tractor engine would be subverted to pulling wind instead of plows. On the other hand, the fan speed could be so low that the radiator cost would be abnormally high, and thus our problem becomes also one of where to content ourselves along the power consumption line.

Consider for the moment a 35-hp. L-head engine. We find, from Equation 8 and using our standards, that the volume of air required per minute for cooling is equal to $35 \times \frac{6000}{30} = 7000$ cu. ft. From the volume curve in Fig. 3, we would require a 20-in. fan driven at 1775 r.p.m. to effect this delivery. The power required would be about 1.5 hp. and the resulting velocity very nearly checks with our suggested standard of 2000 ft. per min. giving a radiator size of 35×0.1, or 3.5 sq. ft. The power for fan drive, in this case, amounts to about 4½ per cent of the engine power, a figure typical of prevailing practice.

Fan Power Requirements

Suppose, however, that we had chosen a 22-in. fan in place of the 20-in. fan. From Fig. 4, showing power consumption of fans of different sizes, each delivering about 7000 cu. ft. of air per min. through a radiator, we find that whereas it requires 1.5 hp. to deliver 7000 cu. ft. of air with a 20-in. fan at 1775 r.p.m., it would require only about 1 hp. for a 22-in. fan to deliver the same volume at 1400 r.p.m. If the fan and the radiator were properly incorporated in the tractor design from the beginning instead of being thrown on somehow or other at the last minute, 0.5 hp. or 0.375 lb. of fuel could be saved for each hour of engine operation. If the

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tractor is operated 1000 hours per year, the saving with fuel at 20 cents per gallon would amount to $12 per year; assuming that this saving could be applied to only 100,000 tractors, it is certainly worth conserving.

In my recommendations for fan sizes, therefore, I have in mind larger fans and slower speeds than those in present average practice, and propose that power consumption for air deliveries shall not be more than 3 per cent of formula rated engine power. Although this is a subject that could be dealt with more intelligently by the fan manufacturer, I believe that the following table showing recommended fan sizes and fan speeds will about meet the suggested power limitation requirements:

Engine Hp.	Fan Hp.	Air Cu. Ft. per Min.	Fan Diam., In.	Fan r.p.m.
20	0.60	4000	18 (Narrow blade)	1600
25	0.75	5000	18	1600
30	0.90	6000	20	1500
35	1.05	7000	22	1400
40	1.20	8000	24	1400
50	1.50	10000	26	1400

In making this plea for larger fan sizes and slower speeds, I have in mind not only economy of operation, but also alleviation of a number of annoyances such as lubrication, bearing troubles, belt slippage and breakages—all too familiar, and all incident to high fan speeds and overloading.

While I have touched on the fan question only lightly, it is one that, as applied particularly to the tractor industry, should be well worth the attention and study of our fan manufacturers and I hope that the leaders will be far-sighted enough, in uncovering their trade data, to give us full and open information on the subject.

I trust I have been able to demonstrate that cooling troubles are not always the fault of the radiator, and if I have convinced the members of the Section of the desirability of adopting some degree of standardization of the essentials of cooling, I shall feel satisfied.

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LUBRICATION OF TRACTOR ENGINES

By W G CLARK*

Lubrication is one of the most important factors entering into tractor operation. Fully 75 per cent of tractor field troubles can be directly or indirectly attributed to faulty lubrication.

Lubrication in general covers such a broad field that a paper of this kind must necessarily be limited in scope; I shall touch, therefore, only upon that part of the subject that has to do with the engine, dwelling especially upon the relation of lubrication to carburetion and ignition and upon the importance of proper oil selection. The latter subject, that of proper oil selection, offers an opportunity for more extensive experimental work than has been done in the past, because the lack of data on this subject and the neglect of manufacturers to recognize and emphasize its importance are responsible for a large part of field lubrication difficulties.

In designing an engine the method of oiling should be determined by the type and by its class of service. Unfortunately, the deficiencies of engine design and of fuel vaporizing methods often necessitate the selection of an oiling system which may not be the best for the particular type of engine on which the designer is thus forced to use it. This is particularly true of some horizontal engines, in which the fuel is introduced in a comparatively raw state and which rely largely on compression for vaporization. Under such conditions vaporization can at best be only partial, and the use of anything but a non-circulating system is prohibited because of the certainty of oil dilution.

TYPES OF OILING SYSTEMS

There are many systems and combinations of systems used in tractor engine lubrication. I do not intend to enter into a discussion of oiling systems in general, but do wish to touch briefly on a few of them.

The old-time wick oiler, which was used successfully on some of the first engines, has become obsolete with the advent of heavy fuels and modern oiling methods. It

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has been supplanted in the horizontal engines by the self-contained single-flow force feed and the external unit mechanical oiler. Naturally the horizontal engine is not adaptable to the splash system, but if it were the methods of carbureting low-grade fuels found on some of these engines would prohibit its use. Since the oil is used only once, a heavy oil consumption must be expected. Yet when proper carburetion methods are employed these mechanical oilers and non-circulating systems are as efficient as any other. Some vertical engines make use of mechanical oilers for similar reasons, which to me seems attempting a cure without getting at the cause of the difficulty.

The circulating force feed, the splash, and various combinations of these, are the systems by far most commonly used in vertical tractor engines. The splash system is especially easy to regulate, easy for the average operator to understand and care for, and when unhindered by fuel troubles, is economical and well deserving of its present wide use.

Selection of Lubricant

The selection of an oil, like the selection of an oiling system, ought to be determined by the type of engine and the service required; yet we find thousands of tractor operators using any oil sold as a "motor oil" because they think that if they pay a big price for a so-called motor oil it must be all right. I know of no greater mistake in the tractor field today than this common belief that any oil is good oil if the price is high enough or if it is bought in a sealed tin. The farmer is not altogether to blame, because there are no practical tests he can apply to determine the suitability of an oil for his engine, unless he actually tries it, which often proves costly. Too little attention has been given by the tractor manufacturer to finding out the best oils for his engine and insisting on their use. This should be done thoroughly and accurately by every tractor maker if only to insure the satisfactory performance of his product. I know of many tractors that have been unjustly condemned simply because the service man in delivering the machine failed to impress on the purchaser the importance of correct oil and care.

The increasing use of low-grade fuels has drawn more attention to the oil question, and we now find that some of the tractor makers are devoting more space in their instruction books to this long-neglected but important factor. Some of them were forced to do so as soon as they

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commenced to use low-grade fuels. A horizontal engine in which the fuel is not thoroughly vaporized before admission to the cylinders must use a very heavy-bodied oil to prevent immediate and excessive wear, because the dilution of the lubricant by the ever-present raw fuel lessens its lubricating qualities so rapidly that such an oil is required to furnish lubrication after its partial dilution. The fact that kerosene acts as a solvent of oil more readily than gasoline accounts partly for the fact that with a heavy-bodied oil one of these engines will actually run better on kerosene than on gasoline. The thick heavy oil is not cut by the gasoline, thus giving greater piston drag and less complete combustion of surplus oil, and often resulting in greater carbon formation and less power on gasoline.

To illustrate this I will cite a case that came under our observation some time ago. A firm manufacturing a large single-cylinder horizontal engine stated enthusiastically that the engine was developing considerably more power on kerosene than on gasoline, even when the gasoline was unheated. We investigated and found that such was a fact as long as the very heavy oil was used, necessitated by the defective carburetion of the kerosene. However, when a lighter and more suitable oil was used for gasoline the situation was reversed, which serves to show how easily one can be led into erroneous conclusions because of the close relation of carburetion to lubrication.

The selection of suitable oils for any given type of engine should be made after careful laboratory tests, supplemented by equally careful field tests. The right kind of laboratory tests require considerable apparatus and are costly, so that certain recent developments in physical chemistry, of means by which the performance of any oil can be accurately predicted, will do much, if adopted, toward eliminating some of our lubrication difficulties.

Tests of Lubricating Oils

In 1913 and 1914, while with the experimental department of the Emerson-Brantingham Company, E. R. Greer and I conducted some tests on lubricating oils by a new method, which yielded some interesting results.

These tests were not intended as a scientific investigation of lubricating oils for gas engines in general, because they were confined to one engine and one speed, so that the results, while accurate in method, are applicable spe-

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cifically only insofar as that engine is concerned. The following description of apparatus and procedure is a partial excerpt from my report made at the time, for which I am indebted to A. Krieg, chief engineer of the Emerson-Brantingham Company:

"The engine used in these tests was a standard stock job, 5 by 7 in., four-cylinder, four-stroke cycle, vertical, designed for heavy-duty tractor work. It was placed in the laboratory in September, 1913, and used for experimental work until the middle of December. By that time all the bearing surfaces had been thoroughly worked in and the friction of the moving parts had become practically constant. The engine was thoroughly cleaned and scraped and the connecting-rods, valves and timing, checked over and put in the best of condition. Since the success of any comparative test depends upon the exact duplication of every condition for each successive test, it was considered necessary to remove the water pump and governor; the water pump because it was mounted on the crankshaft with an adjustable stuffing box, which was likely to vary in adjustment and so change the friction of the engine. The governor was replaced by a rod, which locked the throttle wide open where it remained throughout the duration of all tests.

"The oiling system was of the pump feed, circulating splash type, in which the oil is drawn by a cam-driven plunger pump from the oil reservoir and returned to the crankcase. Thermometer wells were placed in the oil reservoir, intake manifold and in the engine housing between cylinders 2 and 3, projecting down into the crankcase.

"The cooling was accomplished by an electric-motor driven gear force pump circulating the water from a large supply tank and cooling screen or through an open 52-gal. barrel as desired. By an arrangement of valves and piping it was possible to control the water temperature and regulate the flow. Two pressure gages in the cooling system also made it possible to maintain the same rate of flow throughout all the tests. Thermometer wells were placed both in the inlet and outlet of the cooling system, and oil baths were used for the thermometers in all cases.

"The fuel used was 57-deg. motor spirit, purchased all in one lot to insure uniformity. It was fed to the carbureter by gravity from a 20-gal. tank with provision for measuring the fuel consumption accurately.

"The dynamometer was a Diehl electric cradle-type machine with which could be measured small variations

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in load ranging from 0.0038 hp. at 100 r.p.m. to 0.057 hp. at 1500 r.p.m.

"The speed was obtained from a Schuchardt & Schutte tachometer driven from the dynamometer shaft and checked by a Veeder counter on the same shaft.

"The actual testing period for each oil was about 10¾ hr., and was divided into three parts, namely: Cold Friction Test, Power Test and Hot Friction Test, run in the order given.

"After each oil was tested the engine was subjected to a thorough cleaning to remove all traces of the oil. The oil reservoir was drained, washed out with kerosene and wiped dry. The crankcase was similarly treated and then clean kerosene was put into the oiling system. The engine was then driven for a time by the dynamometer until the kerosene had washed all through the bearings, after which the whole system was again drained and wiped out. This process was repeated with clean kerosene until the kerosene drained from the engine contained no trace of oil. The engine was then allowed to stand several hours to drain thoroughly, after which it was wiped dry before putting in the new oil. This cleaning process was followed after every test.

Cold Friction Test

"The first part, the cold friction test, consisted in measuring the friction horsepower of the engine with the new and unused oil. This was done by driving the engine at its rated speed of 700 r.p.m. with the dynamometer and obtaining the change in friction as the heat of compression warmed the oil and jacket water. Friction power readings were taken every 15 min., together with temperature readings from the crankcase, oil reservoir, water-jacket, intake manifold and room. The 15-min., readings were continued until the friction became practically constant and the temperature changes so small as to be inappreciable.

"This part of the test might have been continued until all the temperatures became constant, but the value of the readings would not have been worth the time required to get them.

"By regulating the speed of the water pump and watching the pressure gages in the water line, it was easy to duplicate the rate of flow of the cooling water for each test, so that any change in the heating or cooling rate in a test could be directly attributed to a thermal difference in the oil itself. This idea of dupli

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cation of every essential condition was kept constantly in mind throughout the tests, and every effort was made to eliminate all variables, so that all changes of friction or fluctuation of any sort might be charged to the lubricating oil.

"The object of the cold friction test was to measure the comparative lubricating values of the oils when new; when considered with the third part, the hot friction test, it forms a good basis for the comparison of the stability of an oil's lubricating value.

Power Test

"The second part, the power test, consisted of a 5-hr. continuous power run with wide-open throttle and maximum load. The 5-hr. run was chosen because it was estimated that when the engine ran under full load for 5 hr. continuously, the oil had been subjected to the same amount of work as in an ordinary day's run in the field. During this part of the test the temperature of the cooling water was maintained about 180 deg. for all the oils, and readings were taken as in the first part as well as those required to measure the fuel consumption.

Hot Friction Test

"At the end of the 5-hr. run the third part of the test was carried out. The cooling system was heated to boiling temperature and the amount and rate of flow of the water regulated to be the same as in the first part. At this point the fuel and ignition were cut off and the engine kept running at the same speed by the dynamometer. Friction, horsepower and temperature readings for the first 20 min. were taken at 2-min. intervals, because of the rapid temperature drop and friction increase during the first few minutes. The intervals between readings were later increased to 15 min. as in the first part, and the run continued until the oil and water cooled down to a point 10 or 15 deg. above the final readings in the first part.

"This part of the test was really the most important of the three as it showed the difference in lubricating value of the various oils after being used in the engine and exposed to heat. This serves as a measure of the stability and lasting qualities of the oil.

"At the end of the third part of the test the oil consumption was measured and a pint of the used oil taken for analysis.

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"Upon the completion of the tests, check runs were made on several of the oils to ascertain if there had been any change in the engine friction. This second series of tests checked so closely with the originals that we were confident the engine friction had not changed.

"There were twelve oils used in the tests, some being especially compounded for the engine and the others being common commercial motor oils such as are used in tractor work. No attempt was made to obtain oils of any special base, so that of the twelve tested, two were from Pennsylvania paraffin crudes, five or six from Mid-Continent crudes and the rest from various semi-asphaltic base crudes.

Results of the Tests

"The data acquired were plotted in diagrams in three groups called the Temperature-Power, Time-Temperature and Time-Power.

"The Temperature-Power group showed the variation in friction horsepower of the engine with the change in temperature during Parts 1 and 3.

"The Time-Temperature group showed the time rate of temperature change of the different oils during Parts 1 and 3.

"The Time-Power group was merely a graphical representation of the entire test in its chronological order, showing the variation in friction and maximum power throughout the tests.

"The temperatures used in plotting these diagrams were those of the water-jacket and crankcase, a different set for each temperature."

The variation in the friction of the engine with the different oils was marked and easily measurable. Two oils of the same gravity and viscosity showed as much as a horsepower difference in friction of the engine, and two oils that gave practically the same results in Part 1 were entirely different after being subjected to heat for 5 hours.

One of the oils, similar in physical specifications to several of the others, but used chiefly for line shafting, was tested just to see how it would compare. It gave good results in Parts 1 and 2, but Part 3 showed that the 5-hr. power test had almost totally destroyed its lubricating qualities.

If there were such a thing in petroleum derivatives as an ideal oil, its temperature-power diagram would be a horizontal line; in other words, its lubricating value

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would be the same at all temperatures within the limits of its use in engines. Those oils whose temperature-power diagrams had the least slope, proved to be the better oils.

In the Time-Temperature group the general slope of all the curves was practically the same, although the maximum and minimum points were different. Since the slope of these curves is a measure of the heating and cooling rates during the various tests, the uniformity of slope indicates that the rate of circulation of the cooling water was uniform. These diagrams might also be used to indicate the thermal capacities or insulating qualities of the different oils, although sufficient data were lacking to warrant any such conclusion.

Factors Determining Lubricating Efficiency

The comparison of the oils was based on three factors: Lubricating efficiency, oil consumption, and fixed carbon and insoluble content. Lubricating efficiency was subdivided into three parts: First, average maximum power delivered by the engine with each oil; second, the change in friction horsepower per degree change in temperature; and, third, the average friction between the temperature of 90 and 200 deg. By giving the three separate parts of lubricating efficiency equal weight with oil consumption, fixed carbon and insoluble content, lubricating efficiency as a whole comprises three-fifths of the total comparative rating of each oil, which seems a fair proportion. I will not detail further the methods of scoring used in these tests. The complete report and all the data were printed in *Internal Combustion Engineering*, May 27 and June 3, 1914, published in London.

Chemical and Physical Tests

Samples of each oil were taken both before and after testing in the engine and were analyzed by a chemist. A careful study of the properties of these oils in conjunction with the results of the tests failed to reveal any distinct relation between the physical properties of an oil and its value as a lubricant. The nearest approach to such a relation seemed to lie in the fixed carbon and insoluble content, as the oils that gave the best results had the least amount of these elements. However, we did not feel justified at that time in drawing any such conclusion, but recent developments in physical chemistry of oils indicate that our beliefs were not unfounded.

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These tests proved to us that the performance of an oil in an engine cannot be predicted accurately by the usual physical tests, as given in ordinary oil specifications. Of course, the viscosity and cold-test properties must be suitable for the service intended, but in themselves they are no basis for judging an oil's utility.

Actual laboratory and field tests are the best gages of an oil's suitability for an engine and will continue to be, until some newer and simpler method is worked out. Such a method, which is chiefly a chemical one, was described in a paper by C. W. Stratford entitled, Standardized Specifications for Lubricating Oils, which appeared in the February issue of THE JOURNAL,* I heartily recommend a careful reading of Mr. Stratford's paper because it seems to offer a simple and logical solution of many present problems in oil testing. Mr. Stratford corroborates our conclusions regarding the inefficiency of the customary physical tests as criteria of lubricating suitability, and he also emphasizes the importance of a minimum of carbonaceous insolubles in oils.

Importance of Clean Oil

Proper care of the lubricating system can be summed up briefly in three words: Keep it clean! It seems to be difficult to impress upon the average operator the fact that lubricating oil wears out and gets dirty, even though protected from external dirt. The average operator forgets that the tractor oil is subjected to six or seven times the service that an automobile oil gets in the same length of time, and as a result he neglects the more frequent cleaning and replacing required. The best oil that can be obtained will wear out and become gritty with use and must be thoroughly cleaned out in order to protect the engine bearings.

Putting fresh, clean oil into a dirty crankcase with worn-out oil is a waste of good oil; the old contaminates the new in a short time. The infrequency of cleaning is augmented by incomplete or poor vaporization methods, because of oil dilution. It is a regrettable fact that too often a tractor operator gages the condition of his oiling system by the quantity of oil in it rather than by the quality and condition of the oil; this accounts for many burnt-out bearings, which were apparently flooded with oil.

Spark-Plug Trouble Caused by Oil

The spark-plug question is worthy of mention in connection with lubrication. There are on the market many

*This paper is reprinted on page 511 of this volume.

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good spark-plugs for every class of service, but some of them have been, and are, unjustly condemned, because of poor selection, improper location, poor carburetion and dirty lubrication.

If the right type of plug is used, if the carburetion is even approximately correct and if the lubricating oil is kept clean, a spark-plug will never foul. That statement is backed by many personal experiences and hundreds of field reports. I know of case after case where spark-plug trouble was totally eliminated by simply washing out a dirty crankcase and using fresh, clean oil.

Author's Conclusion

The problem is one of cooperation among gas-engine engineers, ignition and carbureter makers and oil refiners. To insure lubrication efficiency and freedom from lubrication troubles in the field, the tractor manufacturer must first ascertain the oils suitable for his engine; second, recommend a sufficient number of them to cover all territories, insisting on their use; and third, help to educate tractor owners to the importance of correct lubricating oil and its proper care. This need cause no discrimination as regards oil refiners because it is possible to find several oils from various sources which are equally suitable for any particular engine.

In conclusion I repeat that I believe the field of lubrication offers fully as great an opportunity for research and development as any other in tractor work, and that such research and development must be carried out in conjunction with carburetion, ignition and engine design.

THE DISCUSSION

C. P. Fortner:—The following contains a few statistics in regard to the oil situation at the present time:

"In 1916, when all United States oil fields produced 312,000,000 barrels of crude oil, the demand exceeded the production. In 1917 the entire production was 341,000,000 barrels. This was all consumed and the reserve stock was drawn on for 50,000,000 barrels. During the last two months of the past year, on account of the extremely cold weather, drilling for new production was practically suspended. In one field alone there were 115 drilling rigs set for drilling, of which 80 did not turn a wheel. This means that new production has already been reduced nearly 16 2/3 per cent. It will be impossible to overcome this shortage. It is safe to say these same

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conditions existed throughout Kansas and Oklahoma. As we used 50,000,000 barrels from the reserve stock in 1917, probably with the increased demand on account of the increase in the automobile and tractor business, and also in the steamships now building and to be built, nearly all of which are oil burners, the demand for this year will double the amount drawn from the reserve stock, or 100,000,000 barrels."

However, for this year at least there will be enough fuel in the market for tractor consumption; I cannot say anything about the price, but I can state that there probably will be enough fuel.

The method of distribution is very important. My company is a very large distributer of fuel all over the United States—all over the world at one time—and during the past three years we have been establishing stations, until we now average a station every 20 miles on the railroad lines throughout the country, with the exception of the State of Texas. Further than that, we are rendering a valuable service to the farmer by giving him the benefit of this service. From a study of maps we have worked out rural routes, and at nearly all of our stations, as fast as is possible, the agent is being equipped with a tank wagon—a motor truck wagon, in fact—in order to make deliveries direct to the farmer. At the present time the farm is considered a plant. The farmer has a stationary engine, an automobile, and possibly a tractor, and he is entitled to the service which we have worked out for his benefit. I think this will be appreciated by the tractor builders, as it helps them to sell their product throughout the country. The farmer will get his fuel and his lubricants right at his door.

J. V. MAHONEY:—Several years ago, with the old splash-tank type of tractors, practically all the trouble experienced was due to ignorance on the part of the operator. In one case a tractor was delivered with an engine having a non-circulating splash system. The engine was of the four-cylinder vertical type, the crankcase having one compartment in which the oil was permitted to slop from one end to the other. When the machine was traveling up hill all the oil went into the rear cylinder and when going down hill it was all in the front. On a hilly field the farmer spent about 90 per cent of his time cleaning the front or the rear spark-plugs. The farmer asked the expert who delivered the tractor how often he would have to put oil in the crankcase. "Well," the

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expert said, "not the first year, and only once the second year." This was out in Glasgow, Mont., and I believe the farmer's repair bill in about two weeks amounted to $780 and my time.

R. E. PLIMPTON (M. S. A. E.):—It is believed that we can formulate specifications for lubricating oil. The suggestions in Mr. Stratford's paper appear reasonable, and they are complete and definite. I would like to ask Mr. Clark whether, if the oil refiners and the tractor engineers and others actually using the oils will cooperate, we cannot get some lubricating oil specifications that will be of benefit to everybody, so that we can get the right quality of oil and get it at a fair price.

MR. CLARK:—The procedure outlined in Mr. Stratford's paper appeals to me as a possible means of doing away with the tests used in the past for determining oil suitability. It seems logical, and in my opinion it is all right, although it is new and has not been tried out thoroughly.

MR. FORTNER:—As regards lubricating oil, it seems to me that if the engine requirements continue to increase the oil men will be backed up against the fence. Oil is not phenomenal, and has its limitations. On this proposition the oil men would like to ask for the cooperation of the engine designers. Lubrication troubles that we are having might be due to faulty engine construction. It is a fact that fuels are necessarily heavier, that the requirements are greater, and that we are contending with something that causes the rapid dissolution of the oil with which it comes in contact. Under certain conditions, after the oil has rested, it will regain its viscosity, but ordinarily the extreme heat and contact with the fuel thin the lubricant rapidly.

Two oils from different fields, subjected to the same physical tests in the same engine under the same conditions, may give altogether different results. A standard lubricant therefore cannot be manufactured unless the base from which the lubricant is taken is known. That is something absolute—that we know and can all agree on. We should know the base of the crude in order to arrive at the correct base for the lubricant for tractor or automobile engines of any description.

P. J. DASEY:—Experiments carried out in our experimental laboratory with fuels and lubricating oils indicate to a certain degree the end in engine design to which we must work if more successful operation and service to the customer are to be reached.

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Gasoline as Fuel

Gasoline varies in nature as much as the men who make it. Some refiners make gasoline with a fairly low initial boiling point and a low end point in the distillation range; this is called good or high-test gasoline. Others make a product of heavy material blended with casing-head gasoline, the latter a very light product, in order to make starting easy; in many cases the heavy material is not suitable for use in gasoline engines and condensation results, with loss of fuel and dilution of lubricant.

Other refiners make a product consisting entirely of heavy material which is not easy to start on and also brings about loss of fuel and dilution of lubricant.

The first kind, high test, is really the only gasoline fit to use in an engine not provided with means for heating the gasoline in order to turn it into gas. Its use insures that condensation will not take place and that the lubricant will do its work as intended, assuming that the lubricant is of the proper quality for the particular type of engine being used.

It is safe to say that in the majority of cases it is better to use some kind of heating device in order to assist gasification of the heavy ends of the fuel, even though there is a slight loss in volumetric efficiency. The reasons are that greater economy is obtained and danger of bearing trouble is reduced owing to the absence of lubricant dilution and there being no necessity for changing the oil as often as would otherwise be required.

Kerosene As Fuel

Kerosene is heavier than gasoline, and, puond for pound, gives more heat units than gasoline, but it is far more difficult to use. It also varies in composition somewhat the same as gasoline, although to a less degree, for the reason that laws have been passed to keep refiners within certain limits as to the light end content, while the necessity for having an oil that will not smoke when used in lamps has caused the refiners to keep the heavy ends as well within certain limits, giving the user an oil well cleaned and practically water white in color.

The average engine of today will not handle kerosene as efficiently as gasoline because of the inherent qualities of these fuels, which are so little understood.

High test gasoline, deg. fahr	302
Good gasoline, deg. fahr	410
Common gasoline, deg. fahr	430
Poor gasoline, deg. fahr	450

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The end points of different grades of gasoline should not be above the temperatures noted.

Average gasolines have end points between 410 and 430 deg. fahr.

Kerosene varies from 475 to 500 deg. fahr. end points and has an initial boiling point of from 265 to 300 deg. fahr.

From this it will be seen that a gasoline with a distillation range from 120 to 410 deg. fahr. has considerable kerosene content—say all of that part of it distilling over between 302 and 410 deg. fahr., which will average around 20 to 50 per cent.

On the other hand, kerosene can be said to contain a considerable proportion of gasoline, if one chooses to look at it that way, as all of it distilling over below 410 deg. can be classed as gasoline.

Personally, in my research work, I take kerosene as I find it and call it kerosene, and likewise I consider as unadulterated gasoline only that portion of commercial gasoline which distills over below the point where kerosene begins; for laboratory purposes I have set that point at 302 deg. fahr.

In the tractor field as in the automobile field we are forced to take gasoline as we find it for sale on the market. It is safe to say that about a half-and-half mixture is being sold, although the kerosene half is the lightest end of the kerosene, and is handled more easily than a mixture of 50 per cent high test gasoline and 50 per cent kerosene would be for the reason that the former does not contain the heavy material which causes so much trouble.

Splash Feed Lubrication

The lubricant used in splash feed engines must be fluid enough, even when cold, to be readily distributed, yet have sufficient body when hot to keep the wearing surfaces apart under maximum load working conditions. The oil is forced to the bearings solely by gravity, and, under such conditions, it must be very thin to work in between the surfaces. When it is considered that the load on the bearings of a tractor engine is practically maximum at all times, it is remarkable that any great service is obtainable from such as are splash fed, but we know that they do give good service in many instances.

With the oil in a thinned-out condition from heating and then cut or diluted with kerosene, the condition is

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far worse and damage more likely to result than in an engine using gasoline, because the body of the oil is reduced in resistance and must naturally allow metal-to-metal contact at a much earlier load stage than when the oil is undiluted.

If to this condition of reduced efficiency in the oil is added that of fine particles of silicon or sand taken into the cylinders and washed down into the crankcase, also the grit which gets in through the breather tube, we have a condition that is really serious. The sand is bad enough in new oil but its cutting action does not get into full force until the oil has become diluted. The greater the dilution the greater the cutting effect will be; hence the vital importance of keeping out the fine silicon particles and preventing dilution of lubricant.

Pressure Feed Lubrication

In heavy-duty work pressure feed to all wearing parts is of great importance. In the first place, it permits the use of a heavier-bodied oil, because the oil is moved by pressure to the necessary points. The heavier oil will stand up under heavier working pressures. The oil feed increases as the speed increases, and in case of dilution the heavy oil can stand more than the thinner oil without reaching the danger point.

Dust or silicon is not as effective in heavy oil as in light, thin oil. I do not mean to say that dust-laden oil is not to be considered a poor lubricant in the case of the pressure-fed engine. On the contrary, I am convinced that the first tractor engine requirement is to completely eliminate the ingress of dust or sand, and unless this is done efficiently some other prime mover will make its appearance in the near future and eventually displace the internal-combustion engine in the tractor field. I feel sure, however, that in this great industry there are brains sufficient to solve this matter in short order, once its importance is realized. In fact it is well on the way to solution at the present time.

Governor Control of Engines

While governors might not seem to have anything to do with lubrication, upon close analysis the two subjects are found to be intimately connected.

There are two sources of heat in an engine, the burning of the charge in the cylinder and friction. All oils have limits of temperature beyond which it is not safe to heat them, if best results are to be obtained. Let

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us assume that an oil is working well in a four-stroke cycle engine, wide-open throttle, running at 1000 r.p.m. (500 explosions per minute), and consider the explosive and frictional heat generated at that speed. Any increase in engine speed, with load, will necessarily result in considerably more heat from the burning charges, giving a larger percentage of frictional heat because the frictional load increases much more rapidly above 1000 r.p.m. than it does below that point; especially is this true if radiation has been figured on the basis of 1000 r.p.m., maximum working load. Therefore, to my mind it is of prime importance to regulate or govern positively the engine speed not to exceed 1000 piston ft. per min., and provide the necessary factor of safety in radiation.

I have recently made some tests of oil, the results of which, as noted below, are accurate within a fraction of one per cent.

New Oil Test

The contents of a well-known lubricating oil were as follows:

Material distilling up to end point of kerosene distillate, per cent	10.50
Lubricating distillate, per cent	86.75
Coke, per cent	2.75
Total, per cent	100.00

The same oil, used with high test gasoline fuel for 20 hours, under full load at 1000 r.p.m., was of the following composition:

Material distilling up to end point of kerosene distillate, per cent	0.00
Lubricating distillate, per cent	95.50
Coke, per cent	4.50
Total, per cent	100.00

This shows that at the end of 20 hours' run, all of the light material had disappeared and the coke had increased from 2.75 to 4.50 per cent, that is 1.75 per cent.

The test was run with high-grade gasoline to prevent dilution, and it shows not only no dilution, but that the light ends of the lubricating oil itself are missing, having been used up as fuel.

The next test was with a similar new oil, after it had

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been used for one week, under full load conditions, with kerosene for fuel:

Material distilling over up to end point of kerosene distillate, per cent	40.50
Lubricating distillate, very light, per cent	55.00
Coke, per cent	4.50
Total, per cent	100.00

This test shows 40.5 per cent dilution, a net gain of 30 per cent above what the new oil contained in light material, and does not take into consideration any part of that light end that undoubtedly was consumed, as in the first test mentioned. It does show 40.5 per cent light material, which is really a fuel, 55 per cent of a lubricating distillate too light for any purpose except high-speed spindle work and an increase in the coke (carbon).

The separation of the carbon in the form of coke removes the large percentage of real lubricant or body of the oil.

It will be noted that in the oil used one week the coke content is larger than in the unused oil, where it is but 2.75 per cent. This increase is undoubtedly due to a partial breaking down or decomposition of the lighter portions, the light gases being burned and the heavier ends containing the tar carried or washed down into the crankcase.

Later, a series of tests will be made with different oils to determine how long it is advisable to allow an engine to operate with the same oil, using gasoline and kerosene as fuels. These determinations appear to be very important owing to the fact that a certain percentage of dilution must be figured on in all kerosene-burning engines, especially those of the vertical type, until the use of kerosene fuel has reached a more developed stage. Undoubtedly some engines operate with less dilution than others, but thus far I have not received a single sample of oil that does not show dilution. These have been from all types and sizes of engines.

At a recent meeting of the American Society of Agricultural Engineers it was stated in a paper read by a professor of engineering of a large western agricultural college that an analysis of the "carbon" taken from the cylinders of engines in use in the field showed:

10 per cent which would dissolve in gasoline.

10 per cent which would dissolve in chloroform.

80 per cent silicon.

The first thing to consider in the solution of the tractor engine lubrication problem is to strain the air enter-

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ing the carbureter and breather tube so that it will be as nearly 100 per cent clean as possible. That will eliminate the cause of a large part of lubrication trouble. The next question is to govern the engine speed properly. Pressure oil feed to all wearing surfaces should be employed; the tractor manufacturer should use and insist on customers using the best lubricating oil obtainable, changing it as frequently as is determined necessary by positive test. If fuel-oil engines are developed to a higher point of efficiency to avoid dilution only minor difficulties will remain.

E. A. LA SCHUM (A. S. A. E.):—Lubrication, or rather lack of lubrication, we believe is one of the things that caused us much trouble and expense in operating the motor vehicle equipment of the American Express Company. This condition has developed more especially since it became necessary for us to use the heavier fuels. We were seriously troubled with oil dilution owing to gasoline working past the pistons into the crankcase, and, of course, this was more pronounced during the winter months. We were buying cylinder oil from various refiners or jobbers, believing that we were getting the best oil produced, but after much trouble with burnt-out connecting-rod bearings, scored cylinders and pistons, lack of power, difficult starting and waste of fuel, we decided to do a little research work, and therefore purchased some laboratory apparatus, consisting of a standard Saybolt viscosimeter, apparatus for flash and fire tests, and the necessary apparatus for making the cold test. We commenced making tests of various brands of cylinder oil in the early part of the winter of 1917.

A little practice soon convinced us that gravity and color were of no importance to us; that flash and fire figures were of comparatively little importance, but that viscosity was of great importance; of course, we had to provide oil that would pour at the temperatures of winter. We started out with the idea that the oil we were using was not heavy enough for the purpose. We believed that by using the heavy oil we could practically dam the flow of gasoline past the piston and bring about higher compression, more perfect combustion, cleaner valves and hence greater power, all of which goes a long way in bringing about economy of fuel and the elimination of troublesome repairs caused by dilution of oil in the crankcase.

By experiment and observation we have reached the

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conclusion that an oil suitable for our purpose should be about as follows:

Gravity, deg. Baumé	19.3
Flash open cup, deg. fahr., about	370.0
Burn, deg. fahr., about	400.0
Chill at, deg. fahr	0.0
Viscosity at 100 deg. fahr., sec	470.0
Viscosity at 212 deg. fahr., sec	56.0

We are now using oil which meets these specifications and the result has been most gratifying. Burnt-out bearings are a thing of the past; oil dilution does not trouble us; engines are more easily started and drivers report much more snap, which is a very satisfactory condition all around.

I agree with Mr. Clark that the selection of proper lubricating oil offers a big opportunity for more extensive experimental work than has been done in the past, because lack of information on this subject and the neglect of oil refiners to lend assistance have been responsible for many lubrication troubles. The selection of oil should be determined, to a great extent, by the type of engine and service required, but I have found no difficulty in using the same grade of oil on the several types of engines operated by the American Express Company. We have a number of sleeve-valve engines and a great number of poppet-valve engines, also high-speed and low-speed engines. We use the same grade of oil in all types, even in the horizontal engine.

We believe that the performance of oil in an engine can be rather accurately predicted by the physical test as given in the ordinary oil specifications, and that the consumer who operates a large number of motor trucks should make frequent laboratory tests of samples taken from regular stock deliveries to make sure that he is getting the oil specified.

I am satisfied that many operators rely too much on the oil level gage. Spark-plugs will not foul if carburetion is correct and lubricating oil is properly selected and kept clean. I feel sure that maintenance costs can be materially reduced by carefully working out the lubrication problem. I have found this simple and easy to do, and believe that every operator of motor trucks should give the matter immediate attention.

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FUELS FOR TRACTOR ENGINES

By J L Mowry*

It is evident that much time and money have been spent in the enlistment of technical skill to the end that the production of internal-combustion engine fuel may keep pace with the rapidly increasing demand. Let us therefore focus our attention upon engine production and operation—with food for thought drawn from limited laboratory and more extensive field observations.

Oil Refiners

Beyond doubt the oil refiners would have us use a fuel which they can produce in largest possible volume, of the highest possible specific gravity, and lowest volatility. We are using, with a large measure of satisfaction, fuels which have an end-point of 450 deg. fahr. How much farther than this can we go? How much farther will we be asked or *forced* to go?

Accessory Manufacturers

Manufacturers of carburetion devices will warmly welcome the arrival of a fuel of constant characteristics. Their problem has been highly complex—the delivery of a variable fuel under varying load, speed and weather conditions. Much has been accomplished, but an examination of the field of tractors reveals the fact that in the end the adaptations have been made by the engine designers and builders and to them is largely due the present measure of success in burning heavy fuels.

It has been argued that engine and carbureter designers have been working along lines not parallel but so slowly converging that their meeting point is discouragingly far distant. It cannot be denied that most tractor engine builders are too contented with an adapted automobile carbureter. The more efficient utilization of present fuels and the possible use of still heavier oils is more contingent upon engine than upon accessory design.

The spark-plug manufacturers are making every effort to combat the evils of high temperatures and unequal

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expansion of materials. Present international relations may be, in part, blamed for the state of the industry but they should, on the other hand, be a powerful incentive to American producers to invent or discover the necessary dielectric and electrode materials. Spark-plug failures are by far too consistent in our present kerosene engines. Plugs have been in constant use for three years without being cleaned, but probably in such cases some credit must be given to engine design, lubrication and carbureter adjustment. A tractor operator who must replace one to four plugs per day is exceedingly unfortunate.

Without intending to open up at this time the subject of battery vs. magneto ignition, it is my belief that we shall depend upon the latter in tractor work. I have nothing to exact from magneto manufacturers from the standpoint of tractor fuels. We do need, however, a better product from an insulation and dielectric standpoint.

Fuel Needed by Engine Builders

The engine designers and builders are looking for a fuel which will permit the use of standard materials, the combustion of which is not destructive of engine parts, which is not destructive of the lubricant, and which will burn clean.

Our field men and designers are all too familiar with valve failures and partial combustion. The latter, it must be admitted, is in some measure due to quality of lubricants.

It is human nature and habit, born of respect for past achievements and successes, to cling to standard practices. I want to venture the opinion that the present poppet-valve engine is at its height. In so saying, I confess that the substitute is not forthcoming, but far more seemingly insurmountable tasks have been mastered in the past nine months.

The users and operators are the men who finally locate our errors and direct our successes. This class of men demands a fuel which is easily obtainable in all markets, is of constant quality, and is non-destructive of parts and of the lubricant. They do not care what this fuel may be, provided they can get from it reliable power at low cost and small upkeep.

Of many users of tractors only 8 per cent objected to the cost. Of this number some were purchasers of the lowest-priced tractor on the market, and one had been enjoying free service covering both labor and parts. Price objections may therefore be considered lightly.

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Field Observations

The attitude of the user can best be shown by an analysis of returns from 400 users of tractors in the State of Minnesota. To be fair to the manufacturers, all machines older than three years were eliminated as being out of date.

It is to be noted that no distillates were used, likewise no motor spirit. Gasoline and kerosene were used exclusively. Of the total, less than 12 per cent used gasoline except for starting. It is evident, therefore, that the farmer is a user of low-grade fuel.

The average time lost in the field due to engine troubles was 30 minutes, with a range from 15 minutes to 3 hours. This is surely not an unreasonable time for minor field troubles; from this much satisfaction may be derived by designers and builders. The service men can point to such a showing as indicative of their success in making passable operators of new owners within a few days.

It is worthy of note that a man might not advise his neighbors to follow in his footsteps, although satisfied with his own investment. His ego crops out; he considers himself a better mechanic than his neighbors. While only 7 to 8 per cent doubt or deny the success of their purchases, over 40 per cent would not advise Mr. Average Neighbor to make the same investment.

In the light of these reports it is rather difficult to make a case against present tractor fuels. It is to be noted also that the success or failure of a tractor in the hands of a farmer is contingent upon the size of his repair bills. The average age of these three-year tractors was eighteen months, and the average yearly repair bill was $28.05. This should be encouraging to the designers and sales organizations.

There is food for thought for those who would advocate electric starters for tractors, in the average number of days' use per year—50.8. The care of the storage batteries during the idle periods is of utmost importance.

Laboratory Tests

We have been able to pursue investigational work to a very limited extent. The necessity for diluting trained help, and the demand for advice and instruction on automotive machines, due to over-sea conditions, has curtailed work which should have been in progress or completed.

With a four-cylinder removable-head L-type engine

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the following results were obtained, with three kerosene devices:

Fuel, kerosene.
Time, 1 hour.
Water, 190 deg. to boiling.

Series No. 1—Clapper manifold.
Engine speed, 760 r.p.m.—full load.
Result, oil in crankcase increased 0.1 lb.

Series No. 2—Holley vaporizer.
Engine speed, 820 r.p.m. (to get same horsepower).
Result, oil in crankcase increased 0.7 lb.

Series No. 3—Duplex manifold.
Engine speed, 820 r.p.m. (same horsepower).
Result, oil in crankcase increased 0.3 lb.
Engine stopped after 35 minutes, due to preignition.

The chief point of interest to be drawn from these tests is that the "hot-spot" manifold gives more promise than others. This finding is not at variance with what others have disclosed. It is also quite in accord with the present trend of kerosene-burning devices. It is worthy of further study and development.

Possible Fuels

Without again going over well-trodden paths by which tractor fuels may be found or made available in large quantities, the present possibilities are:

1 Cracked heavy oils—too high cost.
2 Casing-head gas—limited in quantity.
3 Alcohol—too high cost.
4 Alcohol and gasoline—which may be mixed under such conditions as to remain homogeneous for some time—too high cost.
5 Synthetic gasoline—still too expensive.
6 Set or fixed gas. This would seem to possess the most promise.

To offset high cost of fuel, engine design and economy may come to the rescue.

Author's Conclusion

While waiting for the further development of substitute fuels, the present article must be utilized. This requires:

1 Better spark-plugs.
2 Manifold development, with the hot-spot principle demanding major attention.
3 Carbureter development with means provided for

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quickly changing to a lighter fuel to take care of overload and throttling conditions.

4 Engine design with special stress upon cooling and elimination of moving parts.

THE DISCUSSION

W. G. CLARK (M. S. A. E.):—During the past three months I have been with a school conducted by a large tractor company. We have covered the tractor belt through the middle and southern states, from Texas to Montana. The reference made to the 8 per cent of farmers who are not fully satisfied, but who are a part of the 40 per cent who would not advise neighbors to buy the same tractor, states the case well. I believe it means the farmer, having bought a tractor which he is not quite sure about, does not want to publish the fact that he has been "stung," but is kind enough not to allow his neighbors to be "stung." However, the situation in Minnesota is better than in other parts of the country. I have come in personal contact with some fifteen to eighteen hundred tractor owners and operators in the past three months, and my point of view has been changed. The situation is really much worse than we imagine. The tractor and accessory manufacturers are to blame.

The solution of the present fuel problem is evidently in the use of low-grade fuel, but, notwithstanding that and the attempt of manufacturers to force this solution upon the public, we find an increasing prejudice against low-grade fuel. I say this advisedly, having the past three months' experience in mind.

Our methods of trying out fuels have involved too much laboratory work, I fear, because certain methods of handling fuels in field operations do not correspond with the methods obtainable with a new and clean engine under laboratory conditions. I do not agree with Professor Mowry that the solution of the problem lies in the hot-spot manifold principle. We receive hundreds of letters every day from engine owners and operators who have so-called kerosene tractors asking if we can guarantee our tractor to burn kerosene. They write, "We have a kerosene tractor that does not burn kerosene."

Two men who had a tractor that was sold to them as a kerosene burner came to me last week. They had used it a year and a half and had burned nothing but gasoline, although it was outfitted with a kerosene equipment. They had written to the manufacturers for information regarding the handling of kerosene, and, much to my

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surprise, the reply was that they should stick to gasoline. After advertising and selling their tractors to burn kerosene they advised the owners to use gasoline! What is the explanation? Evidently after selling it they found that it would not do as expected. If we will do what Mr. Beecroft suggested—sacrifice our prejudices—and cooperate with the accessory men and the engine men, we can improve these engines so that they will work.

Hot-Spot Manifold

A great many complaints have followed the use of the system of burning kerosene with the hot-spot manifold. That is why I disagree with Professor Mowry. Before this past experience I had the opinion he holds now, that there were great possibilities in that method of handling fuels, but my faith in it has been sadly shaken because nine-tenths of our complaints have come from owners employing that system. As Professor Mowry has said, we have been too satisfied with past performances of standard products and too loath to make changes in engine designs which have now proved to be absolutely necessary to handle low-grade fuels.

Kerosene, every one admits, requires some distinct external means to assist its vaporization, and the only means at hand at present is the use of heat. Since heat must be used, it is obvious that the conditions of cooling and temperature control available in the ordinary gasoline engine will not do for the kerosene engine. Those are things which engineers themselves are loath to supply, resulting in the dissatisfaction now prevalent among tractor owners.

I want to state briefly my reasons for losing faith in the hot-spot manifold. I am not talking from a selfish point of view, because we are using carbureters that handle fuels with the hot-spot manifold as well as the others.

The present forms of the hot-spot or duplex manifold are very imperfect for obvious reasons. Using a double or duplex manifold means that the entire charge of fuel and air is heated, and in the forms of existing manifolds there is not one I know of that heats the charge uniformly. Consequently in a multiple-cylinder engine one cylinder does not get a charge exactly like the others. The result is preignition, which is hard to control.

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Extra Weight Objectionable

Another reason is that it is impossible to maintain the same power with kerosene as with gasoline. No farmer that I know of is willing to carry around an engine that has 25 per cent extra weight simply for the satisfaction of burning kerosene. He wants to obtain the most work in the shortest time at the least cost. If he has a tractor that pulls three plows he puts on another to see if it will pull that. If he has a 40-hp. engine and gets only 30 hp. after putting on a hot-spot manifold he will have to carry around the equivalent of 10 hp. that does not render service, and the farmer will not tolerate that. Some tractors must have extra weight to hold them on the ground. The average tractor owner is becoming educated and knows what power to expect from an engine of a given size. That is the main reason why that system of burning kerosene is not practical. We want all the drawbar horsepower possible with a minimum weight.

Another objection to the hot-spot manifold arises from the unequal distribution of the heat, the results of which are cracking inside, dilution of lubricant in the crankcase and deposits in the manifold. We know of many thousands of tractors which are being run on dry gas, and they do not have crankcase dilution, but they are not using the hot-spot manifold. Getting so much addition to the oil through dilution is an indication of incomplete combustion. The obvious remedy is to apply something that will give complete combustion. According to the figures given by Professor Mowry, duplex manifolds do not result in complete combustion, and I think that is an argument against them. We know of several systems of handling kerosene which give complete combustion. We hear much about kerosene and gasoline engine lubricating oils. With perfect combustion I do not believe any change in oil need be made. We have to use heat to vaporize kerosene and when we use heat we have to use more fuel. The least amount of heat should be used in the most economical way possible to get perfect combustion. The solution of the low-grade fuel problem appears to lie in the proper application of heat without the sacrifice of flexibility and economy.

The distinction has not been drawn fine enough between merely running an engine on kerosene and really burning kerosene. Too many manufacturers have been content to say "it will run on kerosene," without being able to guarantee that an engine will burn *all* the kerosene. A gasoline engine can be made to run on kerosene

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if one does not care how much it burns. It is the sale of such tractors that has created prejudice against kerosene as fuel. Kerosene is a good fuel if handled properly. As I have said, the remedy lies in the accessory and engine men getting together.

Prof. Mowry has been a little hard on the spark-plug makers. It is true that a few months ago, owing to the disturbances caused by the war, several lots of bad spark-plugs got out on the market. There are, however, spark-plugs on the market today absolutely satisfactory for handling kerosene. I have seen plugs in service two years and a half which have never been cleaned and have never required cleaning. I think a large part of spark-plug trouble can be located in the crankcase owing to improper arrangement or care of lubrication. Most oils have some good qualities, but the best oil, if not kept clean, is no better than poor oil. We have found, in thousands of cases of engines which are even fairly well lubricated, that if the oil is kept clean spark-plug troubles will be eliminated, provided a good spark-plug is used in the first place. Putting too much stress on the spark-plug question, therefore, is not quite fair.

CHAIRMAN GREER:—Most of the hot-spot manifolds in use today have no heat control. The heat to the manifold is greatest at full load and much lighter at light load. That is the reverse of what is desired.

A. C. BENNETT (M. S. A. E.):—One thing not brought out, perhaps, by Professor Mowry and Mr. Clark is that there is more or less condensation of the kerosene in the manifold with uneven loads, such as are usual under actual working conditions. If a large amount of heat is not used with the hot-spot manifold when operating at light loads, condensation is likely to result. Possibly some improvements can be made in the hot-spot manifold or hot-jacket, as it is commonly called, which would give better results.

Lubrication

MR. CRANDALL:—My experience during the past seven years has been mostly in the field in Montana, Minnesota and Wisconsin, partly in high altitudes and on the prairies. I differ with Mr. Clark on one or two details. One point is lubrication. A heavy-bodied oil must be used when burning kerosene. If the tractor has complete combustion, that is, if the kerosene is vaporized so that there is no smoke from the exhaust, it is still

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necessary to use heavy-bodied oil. I have had experience with that in the mountains on the gumbo tableland where the plowing is very heavy. I have tried to use the ordinary gasoline-engine oil and have found that it cannot be done successfully with kerosene. One reason is that the temperature of the kerosene engine must be higher than that of the gasoline engine, and when the engine is running at full load the cylinders will run dry; I have opened the crankcase on a gasoline engine, put my hand on the cylinder and found it almost dry. Changing to kerosene tractor oil gives better lubrication; that is a recognized fact, and oils for that special purpose are put out.

Gasoline, like kerosene, goes by the piston. I think the best way to completely vaporize the kerosene in order to operate the kerosene engine successfully is to heat the manifold.

Two-Cylinder Engines

My experience has been mostly with two-cylinder engines, the intake and exhaust manifolds being cast together. By night time the exhaust manifold would become very hot, the intake manifold inside of that would be fairly hot and the vaporization of gas would be complete. In the field there is scarcely any smoke after a twelve-hour stretch, but at any time the engine could be loaded up with kerosene so as to emit volumes of smoke. It will work as successfully on kerosene as on gasoline, but it is difficult with a four-cylinder engine to get the intake and exhaust manifold cast so as to heat evenly. I think the thing we should work for now in the four-cylinder engine is to heat the kerosene evenly up through the manifold. To do that it is necessary to have a short manifold.

The two-cylinder opposed engine cannot, of course, be operated successfully that way. It is almost an impossibility to avoid condensation in a long intake pipe. The idea we have got to work toward is to bring the kerosene on the four-cylinder tractor immediately to the point where it is completely vaporized when it goes into the cylinders and not allow condensation in the angles of the intake pipe.

I have done expert work on tractors for about seven years and am positive that the tractor can be brought up to the point where it will burn kerosene as successfully as gasoline.

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G. C. ANDREWS (M. S. A. E.):—Loss of power with kerosene might be due to preheating; in changing from liquid to vapor form some of the explosive force might be lost.

MR. CRANDALL:—I believe Mr. Andrews' point is that in vaporizing gas to a great extent before it enters the cylinder we lose considerable power. I think just the opposite is the result; if the gas is completely vaporized before entering the cylinder there will be a concentrated and clean explosion, whereas the fuel entering in too raw a state has the same effect as too rich a mixture in an automobile. If an engine were started with a mixture too rich or with very raw gas, the explosion would occur all right, but it would not have the force, the combustion would not be instantaneous. Kerosene taken into the cylinder in too raw a state is partly the cause of loss of power instead of the opposite being the case.

MR. COUSE:—Twenty-five per cent more power with 10 per cent less fuel consumption has actually been secured by burning kerosene.

C. S. WHITNEY (A. S. A. E.):—Prof. Mowry made the statement that fifty days of the year were all that the tractor would be used. In replying to a question put to me as to whether a storage battery would freeze, I made the statement that if the owner of the tractor put it away for the winter season he should, of course, take the battery out and put it where it could have attention. When I made that statement at the last meeting, Professor Chase of the University of Nebraska, protested vigorously, contending that the tractor works the year around, cutting feed and doing other work. "You have got to produce something that works twelve months in the year," he said.

PROFESSOR MOWRY:—The figures I cited are Minnesota figures; they are not general over this part of the country. The range is from 21 to about 130 days; two or three owners used the tractors as much as 130 days, running well into the winter.

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PENNSYLVANIA SECTION PAPER

SHOCK ABSORBERS FOR EASY RIDING

By Mark H. Landis*

The various elements that affect easy riding of an automobile were discussed during the past winter by the Pennsylvania Section of the Society. Springs, tires and cushions were treated in considerable detail, and the conclusions reached have thrown some new light on those subjects.

The conclusions of this meeting were, that in order to provide the ideal spring suspension we must have a spring with as low a rate as possible, that is, a "soft" spring, with adequate means for checking its recoil movement without interfering with its compression and without tending to stiffen the spring. Interleaf spring friction was considered undesirable owing to the fact that the damping effects are much less during the recoil movement than they are during the compression movement of a spring. With the friction increased, the rate of a spring naturally must be increased, reducing its shock-absorbing powers.

It was, further, the sense of that meeting that a spring must compress rapidly to absorb road shocks as encountered and it was deemed necessary to control the spring recoil by means of some auxiliary aid to enable the spring to function properly and by so doing secure easy riding.

Spring Suspensions

Before discussing the general subject of shock absorbers as applied to passenger cars and ambulances, let us consider some of the factors that have influenced the design of spring suspensions as used today. Furthermore, we shall consider the various conditions essential to any spring suspension in order to provide a proper shock absorber to function with the spring itself.

*General manager, Landis Engineering and Mfg. Company

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In the early days of the industry the primary consideration was the development of a successful engine to propel the "horseless carriage," and little attention was given to the subject of spring suspension. Springs as used by carriage builders were considered adequate.

The first automobiles had limited speed and were of light weight. Their springs were of light, flexible construction which fulfilled the requirements very satisfactorily. As the art developed greater speed was demanded. This necessitated more powerful engines, and consequently heavier weight throughout all parts of cars.

The change in car design compelled the use of heavier and stiffer springs which in turn reduced the comfort of the passengers. The first step toward relief was the use of large pneumatic tires to replace the old cushion tires and pneumatic bicycle tires. This change provided a flexible connection between the source of power and its point of application and effected a wonderful improvement in riding qualities.

Still greater speed created a demand for a lower center of gravity, with another change in spring specifications. This change brought out various types of suspensions; however, many objectionable features were apparent in them—some springs were too stiff, while others were too flexible. This condition brought about the first demand for shock absorbers; that is, means for preventing the rough riding so common to the early types of cars.

There were other elements which likewise contributed toward riding discomfort, such as rear axles fixed rigidly to the chassis by means of radius rods, which held the axle and spring at a fixed position, all spring recoils acting directly upward. With a rigid rear axle, the car was always started forward with the tire constituting the only flexible connection between the source of power and its point of application. The folly of this design is now appreciated and practically all automobiles now drive through the springs. The springs thus have more freedom of action, improving riding conditions and reducing tire bills.

Unsprung Weight Should Be Minimized

The desirability of reducing the unsprung weight to the minimum has likewise become a recognized fact. The chassis is the anvil to take the blows as given by the axles or unsprung weight through the spring. The axle or unsprung weight is the hammer. The heavier the

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anvil in proportion to the weight of the hammer, the smaller will be the shocks to the car and its occupants.

The unsprung weight has been greatly reduced by the now almost universal practice of making the transmission a part of the powerplant, the use of wire wheels, and of special alloy steels in axles.

Attributes of Ideal Leaf Spring

If we were to discuss the merits of the different spring types, namely, full elliptic, half elliptic, platform, cantilever, etc., we would ultimately come to the conclusion that, whatever type of spring we thought the best, that type must conform to the specifications of the ideal leaf spring: one having such resilient qualities that it responds to and absorbs the slightest shocks and at the same time is of such strength and range of movement that it absorbs maximum shocks.

A car in motion tends to continue in a straight line unless acted upon by an impressed force, caused by the wheels meeting a hole, rut or bump in the road. This impressed force is absorbed by compressing the spring against the force produced by the car traveling in a straight line, the latter force being proportional to the speed of the car and its weight and load. As the impressed force is determined by the unsprung weight, the smaller the unsprung weight the less will be the force tending to compress the springs, or the smaller the shock. It is self-evident that with a low unsprung weight a low rate is possible for the spring, that is, a "softer" spring may be employed.

Action and Reaction

For every action there is an equal and contrary reaction. The action in this case is the result of the shock received by the wheel. While the shock is being absorbed by the spring against the car as a fulcrum, the car will move upward without an unpleasant shock to the occupants. Now, if the spring which has absorbed the shock be unrestrained as regards the force of the reaction, the reaction will cause the spring to recoil violently with a force equal to that of the action or shock less that part of the reaction lost in friction between the spring leaves during recoil. This friction is less during recoil than it is during compression; hence the damping effect secured by interleaf friction during recoil is at the expense of flexibility in the spring during compression. The fulcrum for the recoil of the spring is the road, so it is

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evident that with the car already moving in an upward direction in harmony with the recoil, sufficient momentum in an upward direction is always imparted to pitch the car beyond the normal height of body or chassis as determined by the static deflection of the spring. The

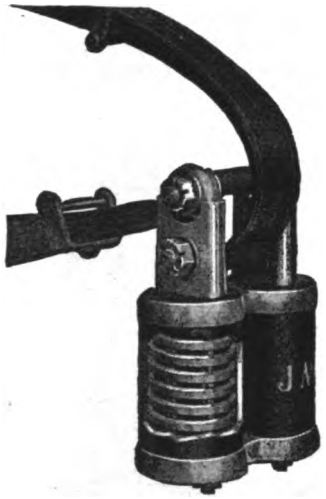

FIG. 1—HELICAL SPRING SHOCK ABSORBER
J. M., Model Z, Twin Tube Type

result is an oscillation of the spring which pitches passengers about and lifts axles and wheels from the road. The lifting of the wheels reduces the adhesion between tires and road. Sometimes this adhesion is entirely destroyed allowing the rear or driving wheels to increase in speed beyond that corresponding to the forward motion of the car. When the tire again grips the road it abrades or rips the rubber from the tread, greatly reducing mileage and life of tires.

The multileaf spring has been offered as the solution of the spring suspension problem, interleaf spring friction being supposed to produce a slow-moving spring that will absorb shocks to such an extent that the reaction will be reduced to the minimum. If sufficient compression friction is introduced the spring approaches the condition of a rigid connection between the chassis and axle, in which case the ability of the springs to act as sensitive shock absorbers is destroyed and reliance must be placed on the tires to do the work of the springs.

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Tires Shock Absorbers for Unsprung Weight

The only function of tires should be to provide an elastic connection between the source of power and the point of application and to act as shock absorbers for the unsprung weight of the car; that is, to reduce the pounding action between the road and axles or unsprung weight. This fact is being recognized today by engineers in the adoption of 8, 10 and 12-in. pneumatic tires for trucks. The extra tire cost is more than saved by the reduction in repair bills.

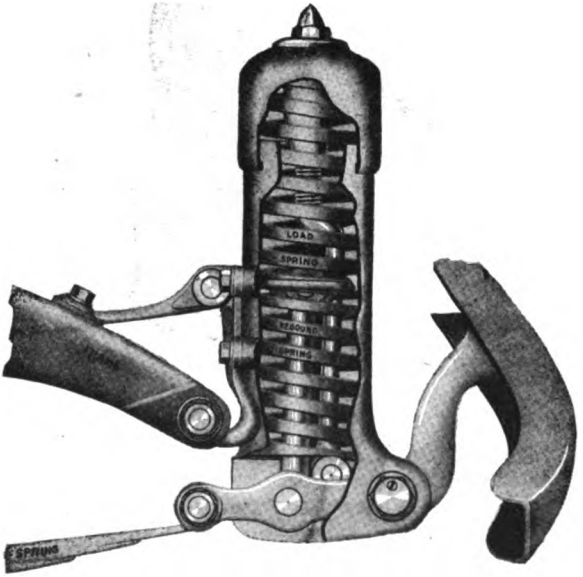

FIG. 2—HELICAL SPRING SHOCK ABSORBER WITH RECOIL SPRING U. S. E. Type Combined with Bumper

At times there is a tendency for a car to leap while running over fairly smooth roads. This leaping is caused by the spring being too resilient during its recoil movements. All leaf springs such as are universally used have considerable interleaf friction. Physical tests show that there is more friction during compression than during recoil; the hysteresis effect reduces the flexibility and consequently the quality of the spring during compression, to a much greater extent than it reduces the spring resiliency during expansion or recoil. The spring without the interleaf friction should be of sufficient strength to carry the static load.

The negative force or friction acts as an additional

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impressed force to be absorbed, that is, the shock plus friction represents the force that tends to throw the car

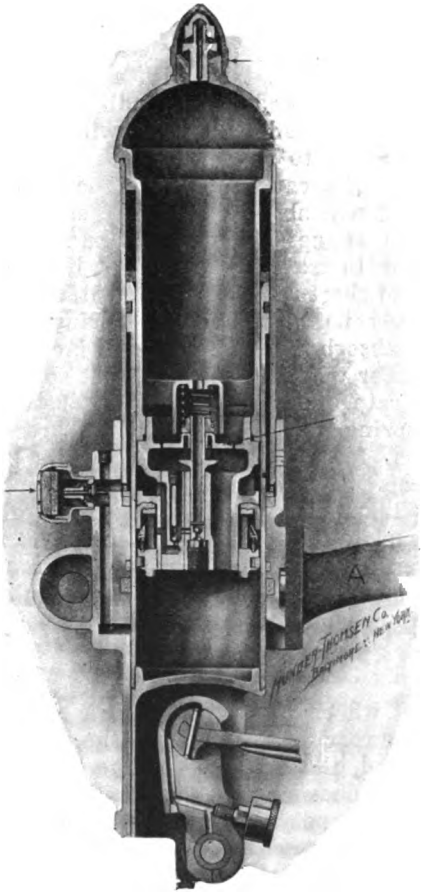

FIG. 3—WESTINGHOUSE AIR SPRING SHOCK ABSORBER

out of its straight line while the shock is being absorbed. When the spring expands, the interleaf friction is much less, and the expansion or recoil is also much more free than it would be if the spring friction were balanced. So it is seen that the combined action of the compression and expansion of the spring, with a much greater inter-

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leaf friction or resistance to its action during compression than during expansion or recoil, starts the car leaping, as both actions of the spring tend to pitch the car upward.

Half-Elliptic Front Springs Predominate

The front springs of 95 per cent of the automobiles are of the half-elliptic type with a comparatively small range of movement and correspondingly high rate. The front springs are made this way owing to the fact that they are the first to meet the shock. The ideal spring for the front of a car should have such range and flexibility that it will absorb the entire shock given to the spring, with range of movement sufficiently great to allow the car to travel in a straight line. If the front springs are of short range of movement, the rate of spring will naturally be high and the springs will lack the capacity to absorb the little jolts and jars of the average road. If a car with short-range front springs, traveling at a high rate of speed, strikes an obstruction in the road, the spring may be completely collapsed during the compression, forcing the car out of its course in an upward leap. If the shock-absorbing capacity of the springs is taken up, the tires must naturally be called upon to complete the absorption of the shock. If the rear springs were designed like the average front springs of an automobile, there would not be enough tire manufacturers to provide the trade with tires.

Causes of Skidding

The front wheels of any automobile should be in contact with the road at all times, for the instant the wheels leave the road, the driver is at the mercy of the rear wheels. The adhesion of the front wheels to the road is to an automobile what the wheel flanges are to a locomotive running on a steel rail. When the front wheel loses its grip on the road the fulcrum by which the car is steered is destroyed.

Such freaks of an automobile as "skidding" and "turning turtle" are directly attributable to the loss of the adhesion between the wheels and road, one of the driving wheels losing its grip on the road owing to excessive spring recoil or a slippery spot in the road. With the engine pulling and only one wheel gripping the road, the speed of the wheel that has lost its grip is suddenly increased. Assuming that the wheel with increased speed suddenly takes hold, and the other wheel starts to slide,

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the high-speed wheel will suddenly sway the car to the opposite side. If at this particular instant the front wheels have not sufficient grip on the road to overcome the sway, the momentum of the rear end of the car will cause it to whirl around. If the rear wheel meets an obstruction sufficiently large to suddenly stop this skidding, the car will "turn turtle."

Summary of Spring Design

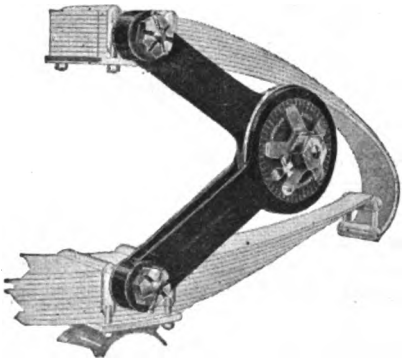

Fig. 4—Hartford Sliding Friction Shock Absorber

The design of a spring suspension involves the following factors:

1 Flexibility of spring mounting.
2 Light weight of car.
3 Low percentage of unsprung weight.
4 Low spring rates.
5 Large range of spring movement.
6 Elimination of interleaf spring friction.
7 Increase of front spring range.
8 Adequate means for control of spring recoils.

We fully believe that if a concerted effort along these general lines is made by the engineers of the automobile industry, a spring suspension far superior to anything produced up to this time can be developed.

Personally, I do not consider an auxiliary aid that controls spring recoil a "shock absorber." The spring itself must be the shock absorber. However, since auxiliary springs and other supplementary devices have become known as shock absorbers, we will use this term in describing all such devices.

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Shock Absorber Classification

Shock absorbers can be classified under the following headings:

1 Auxiliary spring.
   (A) Helical.
   (B) Air.
2 Compression and recoil-damping.
   (A) Sliding friction.
   (B) Flat spring.
   (C) Hydraulic friction.
3 Recoil-damping.
   (A) Recoil strap.
   (B) Sliding friction.
   (C) Hydraulic-pneumatic friction.
   (D) Hydraulic friction.

When we first found that automobiles need some auxiliary means to provide greater comfort in travel we sought relief from stiff springs in the use of auxiliary spring shock absorbers. The auxiliary spring shock absorber "ironed out" the little irregularities in the roads and greatly improved riding conditions, especially at moderate speeds.

However, at higher speeds springs "hit bottom," producing a pounding of the chassis on the axle accompanied by violent recoils. To eliminate this we resorted to the

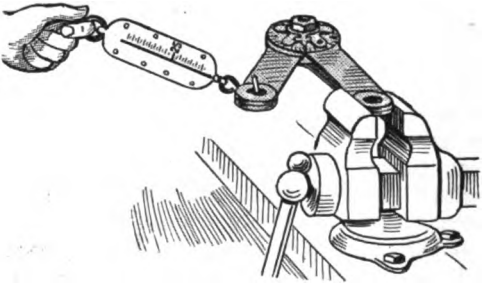

Fig. 5—Adjusting the Sliding Friction Type for Constant Friction

two-way or compression and recoil-damping type of shock absorber.

It next became apparent that we might control violent spring actions by checking only the recoil movement of the spring. This brought into use the recoil-damping type of shock absorber.

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Auxiliary Spring Shock Absorbers

Helical Spring Types

The use of the helical spring as a shock absorber makes it possible to correct the faults of hard or stiff springs found on many automobiles today.

It is a well-known fact that many automobiles with light loads do not ride smoothly when running at low speeds over moderately good roads. This is due to the fact that their springs have not sufficient range of movement and that they are not sensitive enough to absorb all the little jolts. The springs are made stiff enough to provide for the maximum loading but the range of movement provided in the design makes it impossible to use a lower rate spring.

With a helical spring shock absorber, Fig. 1, interposed between the two halves of the spring or between the ends of the spring and the frame, the effective working range of the main and auxiliary springs, as a unit, is greatly increased, making it possible to have ideal shock-absorbing qualities in the spring itself.

However, unless means are provided to control the recoils of this extremely sensitive spring, easy riding conditions will not be provided. Some auxiliary spring

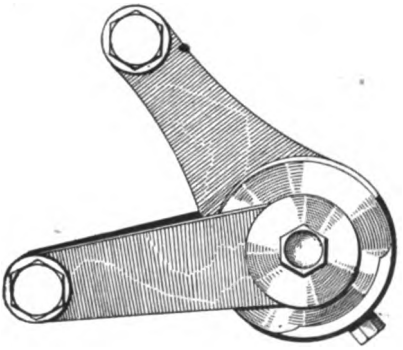

Fig. 6—Connecticut Flat Spring Shock Absorber

shock absorbers provide a recoil spring as a part of the mechanism to arrest the recoils. (See Fig. 2) This recoil spring, to be really effective, must be so strong that it will kill the softening effect of the loading spring. The auxiliary spring has no influence on the recoil effects of the main spring itself other than to make a compound spring of the two, as each spring has a different period.

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At times the difference in period would throw the auxiliary and main springs out of phase and at other times, dependent upon the speed and nature of road traveled, the phases would coincide, tending to increase the effect of recoil.

Air Spring Type

The use of the air-spring shock absorber, Fig. 3, corrects the faults of a hard or stiff spring in the same manner as does the helical spring type. The shock-absorbing qualities of the spring itself are ideal, as the air spring is interposed between the frame and the end of the main spring. It likewise increases the effective spring range of about six inches, giving very soft, flexible action.

The recoil of the air spring is taken care of in the shock-absorber design. However, means are not provided for controlling the recoil of the leaf spring. The criticism of the helical spring likewise applies to the air spring. The air spring and the leaf spring make a compound spring, each having a different period; the difference in period throws the two springs out of phase at times and in phase at other times. When the two springs are in phase, the recoil effect is pronounced; however, the intervals between synchronized phases are much longer than would be the case with the average single unit *ideal* leaf spring; in other words, this device produces a soft, flexible spring unit with a long period. The effect of the recoils may become cumulative, and at continued high speeds the spring movement becomes considerable, so that, unless controlled the recoil of this compound spring of long period will be dangerous.

Compression and Recoil Damping Shock Absorbers

Sliding Friction Type

The sliding friction type of compression and recoil-damping shock absorber was first brought into public use when a remedy was sought to prevent violent spring vibrations, which cause great discomfort in riding.

The effect of "free" and "forced" vibrations causes excessive spring action, making a free soft spring "hit bottom" and then recoil violently. By "free vibration" we mean that vibration which occurs in a spring when it is allowed to expand suddenly after compression. By "forced vibration" we mean that vibration caused by an

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impressed force compressing the spring. If the free and forced vibrations are controlled, the objectionable features of recoil are naturally taken care of.

The sliding friction shock absorber was designed to prevent this excessive spring action. If the spring is not allowed to "hit bottom" the recoil of the spring cannot become excessive; furthermore, when recoil does start, the sliding friction arrests it.

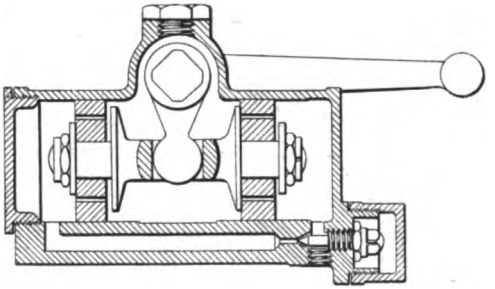

FIG. 7—RENAULT HYDRAULIC SHOCK ABSORBER

Racing Car Shock Absorbers

This type of shock absorber has been used very largely in the automobile racing cars, but not with the idea of securing easy riding. The purpose is to prevent violent spring action which, if not properly controlled, make high speeds impossible.

For the racing car, the pneumatic tire is the only spring or shock-absorbing medium used between the car and the road. If we designed springs to absorb all the road shocks and then prevented the recoils we would be able to secure much more mileage on tires of all racing cars. We would likewise secure better traction and higher speeds. As evidence of the fact that racing cars are so designed that they have no springs or shock-absorbing medium other than the tires, on one racing car that during 1917 had an exceptionally successful season, 5-in. tires were used on a 1600-lb car. If that same car had been equipped with 3½-in. tires, the size recommended for a car of this weight, how long would a set have lasted in a race? In order to keep this car going it was necessary to use tires such as would be recommended for a 5000-lb. car.

The main fault of the sliding friction shock absorber is that it interferes with spring compression; in other words, it produces a stiffer spring, which is undesirable.

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Furthermore, the initial friction, as adjusted for each shock absorber, is a constant at all times, regardless of the speed of spring movements; the resistance is not in proportion to the speed of spring recoils, which is a requisite for every shock absorber or spring recoil controlling device.

Fig. 5 shows the method used for adjusting the friction of the shock absorber. This friction is not a function of the motion of the spring during recoil. This initial friction remains constant whether the recoil is mild or violent.

Flat Spring Type

The flat spring form of shock absorber, Fig. 6, was developed to perform the same functions as the sliding friction type.

Three flat springs, set 120 deg. apart in a casing, are made a part of one arm of the shock absorber and are acted upon and compressed by a triangular cam which rotates on bearings which are a part of the casing. The cam is rigidly connected to the other arm of the shock absorber.

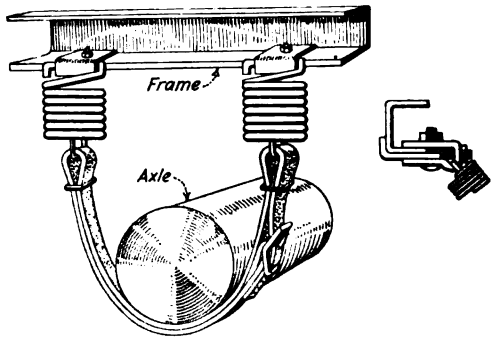

FIG. 8—MESINGER RECOIL STRAP SHOCK ABSORBER

This shock absorber likewise interferes with compression, tending to produce a stiffer spring. However, it does not have any influence on the spring until the spring movements become excessive. As the shock absorber has a neutral point, the effect of working is dependent upon the loading of the car to which it is attached.

Hydraulic Friction Type

The two-way hydraulic shock absorber, Fig. 7, is of the compression and recoil-damping type. It prevents

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violent spring actions by introducing hydraulic resistance which is in proportion to the speed of spring movements. It prevents excessive action at the expense of stiffening the spring on the compression stroke, thereby increasing the impressed force transmitted to the chassis. Another serious objection to this type is the difficulty of retaining the oil in the interior of the casing. The device, to be effective at all times, depends upon the casing being filled completely with oil; otherwise the absence of resistance at the start of spring movements results in a slapping action. This device, when working properly, produces the effects of a slow-moving stiff spring.

Recoil Strap Type

The recoil strap type, Fig. 8, is a very simple shock absorber, designed to prevent excessive recoils. It does not interfere in any way with the compression of the spring, merely reducing the extent of spring recoils.

It has the great disadvantage, however, of tending to lift the axle from the road while arresting spring recoils. It may increase riding comfort, but it certainly cannot fail to increase tire wear and make the danger due to skidding excessive.

Sliding Friction Type

The recoil-damping shock absorber of the sliding friction type, Fig. 9, was designed to snub the recoil actions of a spring. Violent spring recoils are checked when the spring attempts to recoil above the position determined by the static loading of the car.

This device depends on having the strap between the chassis and axle kept under tension at all times. The coil spring acting on the blocks inside of the coiled strap keeps the strap under tension. When the main spring is compressed, the spring coils acting on these blocks follow after and bear against the inside of the strap coil. When recoil starts, the snubbing action tends to arrest the recoil.

This shock absorber tends to interfere with spring compression, owing to the fact that the springs are harnessed down to provide initial tension on the strap. This has the effect of making the spring stiffer. The greatest objection to this type of absorber is that the resistance to recoil is greatest when the spring has the least reactive resistance. In other words, the recoil is arrested, in fact completed, before the spring reaches its normal position, as determined by the static loading, always leav-

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ing the spring rigid at the start of the next compression movement. Furthermore, this device is not self-adjusting, as the snubbing action depends upon the loading of the car, owing to the variation in pressure of the coil springs on the blocks inside the coils. This variation in pressure varies the tension on the strap.

Hydraulic-Pneumatic Friction Type

A shock absorber of the hydraulic-pneumatic friction type is shown in Fig. 10. When the spring is compressed, the valve in the piston opens, allowing oil to flow

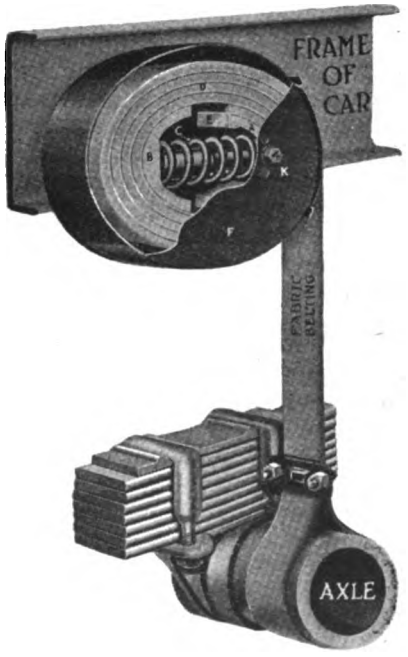

FIG. 9—GABRIEL SNUBBER, OF SLIDING FRICTION TYPE

through the piston. When the spring recoils, the valve in the piston closes and the recoil is met by a hydraulic-pneumatic resistance. The air in the space above the oil level is compressed until the pressure corresponds to the resistance offered by the oil in passing through the by-pass holes in the piston. After the air has been fully compressed, the hydraulic resistance starts, producing a gentle recoil of the spring to the height, as determined by the static load, the resistance to recoil being in direct

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proportion to the speed of spring recoils. If the extent of spring action is slight, the resistance to recoil is slight, as the speed of recoil is a function of resistance; likewise with violent spring recoils the speed of travel is high, with the resistance directly proportional to the speed. Every spring has a fixed period if allowed to oscillate freely; however, the speed of recoil is directly proportional to the amplitude of the spring, if the period is a constant.

This shock absorber has the disadvantage of using a pneumatic resistance to start the recoil stroke. As air is a compressible medium, there will naturally be an increase in speed of spring recoil at the instant recoil starts. However, after the air has been compressed to a certain pressure, dependent upon the size of by-pass in the piston, the resistance to recoil is directly proportional to the speed of the spring recoil, as the oil is an incompressible medium.

We believe that the mechanical construction of this device makes it advisable to provide a pneumatic air cushion, in order to prevent a slap or sudden jolt at the start of the spring recoil.

In a shock absorber of this construction, it is practically impossible to provide an active hydraulic resistance at the instant recoil starts. There is bound to be a cavity above the piston which will permit of a sudden acceleration of the spring at the instant recoil starts. It would be impracticable to have the entire space on the inside of the cylinder filled with oil, as there would be no room for expansion of the oil. The work done in checking or controlling recoil generates heat, and it would be impossible to hold this oil by any packed joint or stuffing-box, when the oil, an incompressible medium, expands, unless an air space were provided.

We are likewise inclined to question the advisability of the piston rod passing through the casing. The stuffing-box requires regular attention to keep it in proper condition. It also becomes necessary to provide special protection for the rod itself to prevent dust and grit from lapping the rod and gland. It is also necessary to have a tight fit between rod and packing in the stuffing-box to prevent oil wastage and to retain maximum pressures. This produces friction negative to spring compression which has the effect of stiffening the spring for short and sensitive vibrations during spring compression. There is likewise a marked tendency for the oil and air to become thoroughly mixed, producing a spongy resisting medium.

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Hydraulic Friction Types

The Houdaille shock absorber, Fig. 11, is a hydraulic device for controlling spring recoils. It introduces no resistance to compression other than the resistance of the paddle journal in the stuffing-box. The recoils of the spring are controlled by the hydraulic resistance of oil passing through by-passes in reciprocating paddles or rotary pistons. The resistance is always in proportion to the speed of spring recoil. An auxiliary oil chamber is provided outside of the casing proper to replenish any

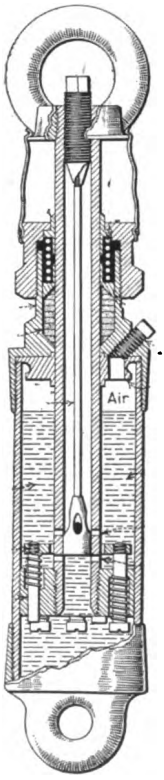

FIG. 10—FLENTJE HYDRAULIC-PNEUMATIC SHOCK ABSORBER

oil waste around the journal or stuffing-box. It is highly essential to have the casing proper filled with oil at all times, otherwise there would be a cavity in the casing, which would produce a "slap" at the start of recoils. With the slightest loss of oil around the journal, the

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paddle produces a suction on the oil reservoir, replenishing the leakage.

The chief objection to this construction is that no space is allowed in the casing for expansion of the oil.

FIG. 11—HOUDAILLE HYDRAULIC FRICTION SHOCK ABSORBER

The heat generated in the oil by the work of controlling the spring recoils will cause the oil to expand, forcing it out around the stuffing-box. There is also the objectionable feature of the inertia of the resisting oil to be considered. The sudden change of direction of the flow of oil in violent and rapid spring action makes it impossible for the oil to follow after the pistons or paddles without agitation or frothing. This agitation produces more resistance to compression than it should and likewise has a tendency to saturate the oil with fine bubbles of air. The mixing of air with oil tends to make the resisting medium spongy instead of incompressible.

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Evolution of the Hydraulic Recoil Check

About seven years ago we attempted to design a shock absorber to control spring recoils. One design succeeded another with many changes and modifications. However, we confined our efforts exclusively to the development of a hydraulic recoil check.

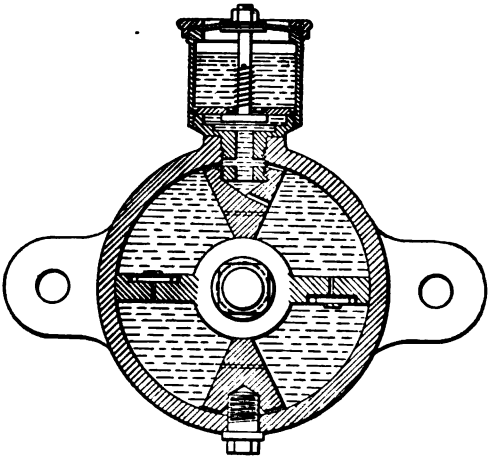

FIG. 12—INTERIOR CONSTRUCTION OF THE HOUDAILLE SHOCK ABSORBER

The first commercial Landis model is shown in Fig. 13, a sectional view representing the action during spring recoil. The rotating arms which are attached to lever *N* and the support arms attached at *W* and *V* are omitted from this view.

Fig. 13 shows the absorber at the start of spring expansion; the oil below the piston *I* in steel cylinder *H* is put under pressure when the spring expansion starts, forcing the oil through the orifice *T*, which is constant in size. By means of the air in chamber *L* in piston rod *D* a pneumatic cushion is provided for changing the direction of action of the spring. The air is compressed to the point of resistance of the orifice *T*, after which compression the resistance is in proportion to the speed of the spring recoil.

Fig. 14 shows the condition at the start of spring compression; the link *C* carries the piston rod upward, first opening the ball valve *P* and next carrying the piston *I* up through the steel cylinder *H*. This action produces a partial vacuum under the piston *I*, which causes the steel cylinder *H* to move upward to the point

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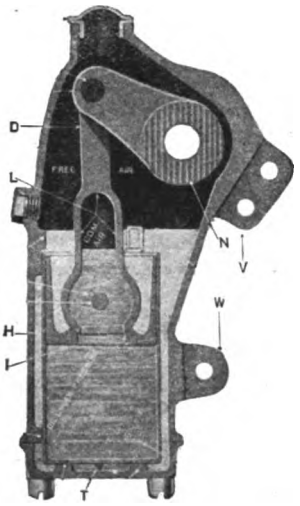

FIG. 13—FIRST COMMERCIAL LANDIS MODEL DURING SPRING RECOIL

determined by the guide lugs *M*. This movement permits of free openings to the interior of cylinder *H*, through the bottom of the cylinder at *R* and through the piston at *P*, enabling the cylinder to fill instantly with oil. It will

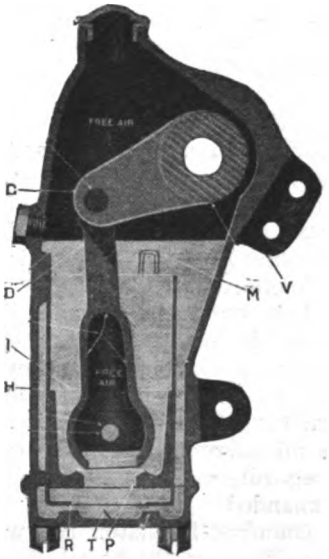

FIG. 14—SAME MODEL AS FIG. 13 AT START OF SPRING COMPRESSION

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be readily seen that this action does not interfere with the resiliency of the spring or in any way retard spring compression.

When spring compression is complete, and spring expansion starts, the piston rod *D* first seats on the seat in piston *I*, closing the valve at *P*, then the steel cylinder seats itself on the seat *V*, closing the space between the bottom of the cylinder and piston, with the exception of orifice *T*, through which the oil is forced by the downward movement of piston *I*.

Although our original experimental designs were purely hydraulic types, this model is in reality a hydraulic-pneumatic type of shock absorber. We found that we were not always able to fill the cylinder *H* with oil, and to provide for the "pound" or "slap" which gave an acceleration at the start of recoil, the air cushion was placed in the connecting-rod.

We were still troubled with oil leakage around the journals of the rotating arms which clamp on the link *C*. We used relief valves to equalize the air pressure on the inside of the casing when the compressed air in the piston expanded after spring recoils. We attempted to pack the joint at journal *N* without increasing frictional resistance to spring compression, but without success. We attributed leakage of oil to the sudden expansion of air in the hollow piston rod, but finally we decided that it would be better to change the direction of flow of the oil after it was discharged from the bottom of the cylinder through the orifice *T*. Instead of discharging the oil all around the steel cylinder we modified the construction of the casing as shown in the sectional view, Fig. 15.

This construction used a by-pass plate *G* which discharged into an offset portion of the casing. During spring recoil this action discharges the oil over by-pass plate *G* causing the discharged oil to establish a current back of the baffle plate *P* along the outside of the casing, the current being deflected by oil deflector *R* discharging downward around the outside of cylinder *B*. When cylinder *B* is raised from its seat at the start of the compression movement, the oil is moving in the proper direction to enter the cylinder at valve *E* and over by-pass plate *G* into the bottom of the cylinder. The cylinder can be filled instantly as the oil current is always travelling in the same direction, regardless of whether the spring is being compressed or expanded.

With the air chamber in piston rod we found small bubbles of air would mix with the oil above the oil line. By means of the valve mechanism at *H* we were able to

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separate the air and oil, allowing the air to escape during spring compression; the separated oil returned to the casing proper through the small pipe *T*, the current of oil as it passed the end of this pipe producing a suction to return the oil.

On the recoil movement of the piston the air in the hollow piston rod is compressed. The wafer valve *H* cuts off outside air communication, and the air above the oil line

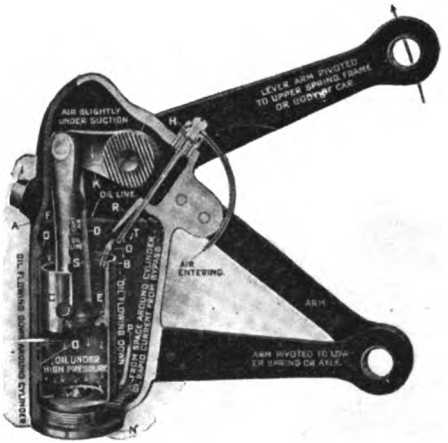

FIG. 15—MODEL SIMILAR TO FIG. 13, BUT WITH CHANGED CASING CONSTRUCTION—SPRING RECOIL POSITION

A—Casing.
B—Floating cylinder.
C—Piston.
D—Hollow connecting rod.
E—Ball valve.
F—Air pipe.
G—Bypass plate.
H—Ring air valve.
I—Oil filling plug.
J—Stationary arms.
K—Short lever
L—Rotating arms.
O—Guide lugs for cylinder.
P—Baffle plate.
R—Oil deflector.
S—Air chamber.

becomes slightly rarefied owing to the compression of air in the hollow piston rod. This produces a suction inward. As the valve *H* is closed, the only place for air to enter is around the journals at *M*. The expansion and contraction of the air in the hollow piston rod in connection with the air valve made it possible to eliminate the breathing action which tended to cause oil leakage at the journals.

Air was replenished to the air chamber by the rush of oil over the slot at the terminal of the small oil pipe *F*, on the face of the piston rod.

This model did not waste oil and handled spring recoils very satisfactorily. However, we were not satisfied with its design and again modified the construction as shown in Fig. 16.

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In this design we dispensed with the air chamber, reduced the lift of ball valve in piston and increased the lift of cylinder *B*. By the elimination of the air chamber we have done away with the breathing action on the inside of the casing, greatly simplifying the construction. We have also been able to introduce a hydraulic resistance immediately at the start of spring recoils. This is made possible by the fact that we do not have to take care of the oil inertia due to sudden changes of direction of oil travel.

The latest design, described above, seems to fulfill the requirements of the shock absorber which we started to produce seven years ago.

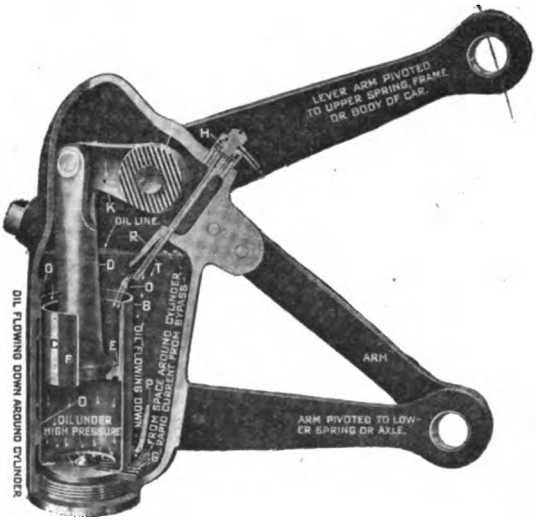

FIG. 16—LATEST MODEL LANDIS SHOCK DIFFUSER WITHOUT AIR CHAMBER—SPRING RECOIL POSITION

A—Casing.
B—Floating cylinder.
C—Piston.
D—Connecting rod.
E—Ball valve.
F—Wrist pin.
G—Bypass plate.
H—Ring air valve.
I—Oil filling plug.
J—Stationary arms.
K—Short lever.
L—Rotating arms.
O—Guide lugs for cylinder.
P—Baffle plate.
R—Oil deflector.

TESTING SHOCK ABSORBERS

Any test mechanism that does not provide for the violent oscillation of the unsprung weight and which does not show the return of the chassis to its normal static load position, as well as the time required to return, is worthless and misleading. A test mechanism as in Fig. 16 shows actual conditions.

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By arranging the spring suspension to travel over an endless-belt highway at predetermined speed, with a reducing motion connection between the recording chart and highway, we can accurately and graphically record the true conditions existing in any spring suspension. We believe it would be for the best interests of all concerned to spend a little time and effort in obtaining some honest technical data on this subject. If future tests do not verify expectations, we should try to find out wherein the difficulty lies.

The shock absorber manufacturers of this country have been laboring under a great handicap, due to the fact that there seems to be great reluctance on the part of automobile manufacturers to consider the use of any auxiliary equipment. The accessory manufacturer must create the demand; he must force the automobile manufacturer to change his car because the public demands it. The day cannot be far distant when we will see marked changes in the designs of automobile spring suspensions, which will result in a combination of satisfactory springs and shock absorbers.

Author's Conclusions

Spring suspension designs should provide sufficient range of movement to take care of all loads from maximum to minimum, using a spring of low rate. To secure easy riding it is folly to attempt to use a spring that has a very small range of movement with a high rate. Shock absorbers should not be expected to improve such conditions. Some life should be put into the spring, and its recoils controlled by a suitable shock absorber.

Summing up the requisites of a shock absorber, its design must be in accordance with the following conditions:

1 No neutral points in its action.
2 High road speeds with comfort and safety.
3 Control of spring recoil.
4 Recoil soft and gentle.
5 Springs not stiffened.
6 Friction a function of recoil, not a constant.
7 Introduction of a recoil time element in proportion to the speed of spring expansion.

THE DISCUSSION

Chairman John W. Watson:—Some automobile designers seem to feel that a recoil check on a spring,

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instead of operating to keep the wheels on the ground, will tend to lift them from the ground. On the other hand, I have satisfied myself by tests that a properly functioning recoil check—one that starts to act immediately upon the reaction of the spring—will prevent the body from ever gaining sufficient upward momentum to pull the axles and wheels off the ground. The only kind of recoil check that can cause the axles and wheels to be lifted off the ground is one that allows the body to gain a certain amount of upward acceleration and then attempts to check this acceleration. That is not, however, the way in which a proper recoil check should function.

MR. LANDIS:—The opposite is true. The spring is held under light tension during its recoil, and there is no tendency for the wheel to leave the road surface. The instant the spring expands beyond its normal position perfect adhesion between tire and road ceases and the tire tends to be lifted from the road. The axle certainly will not be lifted from the road before the spring reaches its normal position. This position is determined by the spring deflection, which is dependent upon the weight of the body and the load in the car.

If we were to put a jack under the body of the car, we would have to lift the car beyond the normal position before the wheels would be off the ground. Up to that point we would be lifting only the body and its load, but the instant the normal point had been passed the weight would be taken from the wheels. We would still have the dead weight of the axle and wheels pulling down or tending to open the spring.

CHAIRMAN WATSON:—The proper recoil check does not act to prevent the separation of the axle and body, but acts merely to prevent their too rapid separation. The function of a recoil check is to prevent the body and axle from separating beyond their normal distance. If the body and axle are not allowed to separate beyond the normal distance, it follows that the spring will always be maintained in a compressed condition. The spring being thus compressed and being resisted by the weight of the body from acting in an upward direction, is therefore constantly tending to force the axles and wheels down and keep them *on* the road.

MR. LANDIS:—Yes, the load would be off the tire but the wheels would still be on the ground. Before the wheels are actually lifted from the road, the car must be well above the static load position. It seems to me

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that the recoil check also keeps the spring under compression during the recoil.

If the wheels pass over a depression, they will naturally tend to follow the road surface. The resistance to opening of the spring offered by the recoil check is proportionate to the speed of the spring movement. With any recoil check mechanism, regardless of the weight in the car, we should not introduce so much resistance as to greatly prolong the period required for the wheels to return to their normal position.

A MEMBER:—When traveling at a high rate of speed, what happens when a series of bumps, such as car tracks, are struck?

R. S. CARTER (M. S. A. E.):—Under such conditions the resultant displacement of the body of the car would depend upon many variable factors such as the ratio of sprung to unsprung weight, the speed, the flexibility and period of both front and rear springs. With certain combinations of these factors it would be most desirable to make use of shock absorbers, while with other combinations undesirable, unless the flexibility and period of the car springs and the action of the shock absorbers were properly coordinated.

A MEMBER:—How should the ideal spring suspension, as mentioned in Mr. Landis' paper, be designed?

MR. LANDIS:—That depends upon the type of spring used under the vehicle, and also upon the speed of the car. If the unsprung weight could be kept down to a minimum, the spring rate could be greatly reduced. I consider that the spring rate should be determined by the weight of the axles, wheels, body, the live load in the car, and the range of spring movement necessary to meet average road conditions. On meeting an obstruction the unsprung weight imparts a certain compressing force to the spring. If the range of spring movement were not sufficiently large, it would be necessary to increase the spring rate.

CHAIRMAN WATSON:—Regardless of the amount to which the spring has been compressed, should not the resistance to the reaction of the spring start at zero and be increased in ratio with the reaction distance?

MR. CARTER:—It starts from zero and builds up. At the first impact of the blow, it should start to absorb the energy imparted by the shock. If it does not start with the shock impact the springs will collapse farther and a much greater resistance will have to be applied in order to check the car springs on the rebound. This

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greater resistance of the shock absorber against recoil will materially affect the recovery of the car springs under certain road conditions where a quick recovery is essential to comfort.

MR. LANDIS:—With a hydraulic type, on the other hand, the resistance decreases with the car speed. That is the trouble I see with most friction recoil devices.

I have seen this tried with very soft flexible springs on cars running at 20 m.p.h.; when passing over ordinary obstructions each spring becomes completely straightened. We tested a car fitted with a suspension designed in accordance with our suggestions, and the springs seemed to be too weak. We controlled the recoil without interference with the compression, and the springs then failed to "hit bottom," even at high speeds.

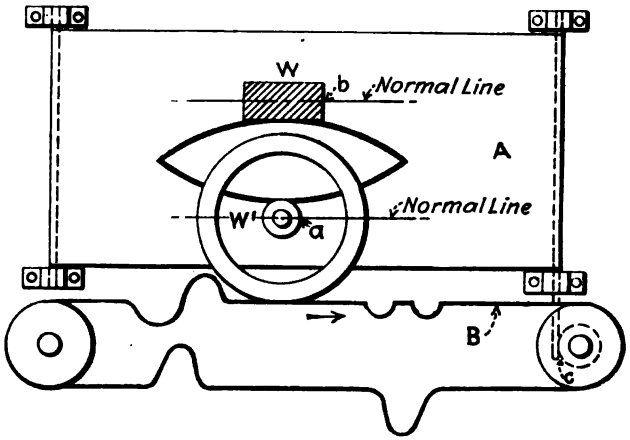

FIG. 17—SUGGESTED AUTOMOBILE SPRING TESTING APPARATUS
Sprung weight, *W*. Unsprung weight, *W'*. Pencil points, *a* and *b*. Record sheet, *A*. Traveling highway, *B*. Worm drive between highway and record sheet, *c*

MR. CARTER:—Most discomfort comes from heavy forced vibration, occasioned when running into holes or ruts; unless shock absorbers are used automobile manufacturers are obliged to supply springs that will meet the worst road conditions even at the sacrifice of comfort. Springs that are not supported during compression by some form of shock absorber are essentially heavier springs which have not as good riding qualities under good road conditions as more flexible springs.

Instead of controlling the recoil alone after the spring has moved considerably under compression, will it not be better to start to absorb the shock energy at the time

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of impact, in view of the fact that a definite number of foot-pounds of energy must be absorbed and the earlier the absorber comes into action the longer the time it will have to absorb this energy?

MR. LANDIS:—Mr. Carter is attempting to absorb the energy with a short movement by using a spring of a variable rate. It is quite true that some of the impact will thus be absorbed, but the tendency to pitch the car from its straight-line travel will be increased while the spring is being compressed, because the spring is of a higher rate. The impressed force acting upon the body is greater; hence the pitching of the car will be greater during compression of the spring.

All that is necessary is to provide a slight resistance so that when the spring is coming back there will be no slap or pound, as there is with a free spring. If the recoil can be controlled immediately at the start, the recoil movement can be overcome with slight resistance much more satisfactorily than if it is allowed to accelerate before attempting to stop it.

MR. CARTER:—Mr. Landis brings out just the point I was approaching, which is that very flexible springs can be used with safety on a car provided they are supplemented and protected against the forced vibrations of impact shocks by the co-action of a properly designed shock absorber.

An advantage of using springs of high flexibility is that (as their variation, per inch of motion, is less than that of stiff springs) their supporting action against the frame of the car is more nearly constant when the wheels of the car are passing over the uneven surfaces of the road. For example, in passing over a mound 2 in. high, a spring of 600 lb. per inch increases its pressure against the frame by 1200 lb. and one of 200 lb. per inch by only 400 lb., and conversely when passing through a depression of 2 in. in the road the heavier spring reduces its support against the frame by 1200 lb. and the flexible spring by only 400 lb. With the flexible spring the variation of support against the sprung mass of the car is only one-third that of the heavier spring.

As the ideal suspension must allow motion of the unsprung parts without variation of reaction of the supporting elements against the supreme mass it is evident that the more nearly this variation is brought to zero the nearer do we approach the desired but unobtainable straight-line travel of the sprung mass.

ERNEST FLENTJE:—Three essentials are necessary to make a car ride flexibly, sensitively and safely: first,

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sensitive cushions; second, long, flexible springs; third, automatic shock absorbers. The main spring of an automobile cannot act quickly enough when going over little holes in the road. In such cases a sensitive cushion will take up oscillation or quiver that the main spring, when carrying a light load, cannot follow quickly enough.

I have found by experience that the half-elliptic spring gives better satisfaction and more sensitive riding qualities with a light load than the three-quarters or full elliptic spring. When only two or three people are riding in a seven-passenger car (having springs designed for a seven-passenger load) the springs will not oscillate freely and the car motion will be choppy.

Importance of Spring Finish

The average automobile springs are not properly finished; the scale from the rolling mill is left on, and in many cases the spring is not shaped properly. A spring not properly scraped acts like a file; such a spring, if taken apart after 4000 to 5000 miles of use, would show many imperfections causing friction and preventing free oscillation.

To overcome lack of sensitiveness in springs, I go over the spring leaves with an emery wheel, taking off the roughness and scale, then I put them on a felt wheel and polish and apply oil and graphite. Then I place a strip of specially prepared spring phosphor bronze between every leaf. This treatment increases the flexibility from 20 to 25 per cent. By this means a seven-passenger car can be made comfortable and very sensitive when two or three persons are riding, but if the car is also equipped with automatic shock absorbers it will be equipped for shocks of either one or one thousand pounds, the latter actually occurring at high speed when meeting road obstructions.

The point has been mentioned that the shock absorber must take solid hold immediately on the recoil. I employed that principle for nearly five years, but found that it was wrong. When we equip a car, the device is attached in the front between the axle and frame, so as to allow from 3 to 3½ in. of movement between the bumps. The rear attachment depends entirely upon the action and the length of the springs, but a piston travel of from 4½ to 6 in. is usually provided.

E. R. Waterman (A. S. A. E.):—Mr. Landis thought the speed or blow had no effect on the initial action of the

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Hartford shock absorber on the downward action of the spring.

MR. LANDIS:—No, I did not say that.

MR. WATERMAN:—The tension on the Hartford shock absorber is about 35 lb. for the average car, and while it is a constant-friction device, the initial shock or blow has a tendency to release the tension momentarily. Then, as the action of the spring becomes weaker, the tension or control of the shock absorber increases proportionately until it offers its maximum of resistance to the spring's vibration. This is important, as the car spring has stored up energy which must be dissipated to prevent a disagreeable recoil, and the absorption of a large measure of this stored-up energy, by the application of the maximum resistance of the shock absorbers, effectively slows up the return of the spring to its normal position.

CHAIRMAN WATSON:—A spring requiring 200 lb. to compress it 1 in. will require 400 lb. to compress it 2 in. If a spring, therefore, has been compressed 4 in., 800 lb. of energy are stored up in that spring. I cannot understand how a resistance of but 35 lb. can have any appreciable or beneficial effect in preventing this energy from tossing the body up beyond the normal position.

It is doubtful if spring manufacturers can ever furnish springs for passenger cars that will give ideal results under the great variety of conditions, without the aid of some controlling device, either as a part of the spring itself or added as an auxiliary. Passenger car springs are designed for full passenger load, but in many cases are operated under much lighter load; consequently, the spring action under the lighter load is more violent, producing greater vibration and rebound (or upthrow), so that some shock-absorbing device is needed for protection of the springs and for riding comfort.

Air Shock Absorbers Give Long Range and Flexibility

RICHARD LIEBAU:—The Westinghouse air spring has a telescopic cylindrical air chamber, allowing an unrestricted movement of the steel spring and a six-inch movement of the air spring. The air springs are inflated to a pressure sufficient to carry the car with a given bdoy load. The air pressure is governed by the weight of the car; for the light car it is from 35 to 40 lb. per sq. in. and for heavier cars from 45 to 90 lb. We do not govern the absorber by the pressure in the spring, but by the position after it has been inflated.

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The oil used has three functions: to seal the air, to lubricate the moving parts, and to operate the little single-acting oil pump for pumping the oil back into the spring to level the oil.

Our experience shows that it is necessary to have steel springs with long range of movement at both ends of the car. Engineers with whom I have come in contact do not seem to consider that blows on the front end of a car have as much to do with the action as those on the rear, but a car equipped with these devices on the rear only will not even give half as comfortable riding as one equipped both on front and rear.

We find that in order to get satisfactory results with our equipment the steel spring under the car must be sufficiently heavy to support the load under normal conditions, and of sufficient strength to work safely on average roads without striking the axle.

The air shock absorber results in a long range of flexible spring action, thus allowing the wheel to follow the inequalities of the road without disturbing the straight-line travel of the body; that is the real reason for the success of the air spring. I do not believe any device has been conceived as yet that will absolutely eliminate all road shocks or rebounds, but we are all working to that end.

JOHN W. BLACKLEDGE (A. S. A. E.):—A spring cannot be made with the correct strength for both a 1500-lb. and a 50-lb. load. Spring-type shock absorbers take care of variations in load and adapt themselves to any load. Their coils, being softer and more resilient than the main spring, work faster than this and absorb the thousands of small vibrations before they are carried to the body of the car. Manufacturers of the type of shock absorber known as the recoil check state that the spring type does not resist the upthrow of a car. In this they are mistaken.

The two following illustrations will serve to explain the manner in which the spring type helps to absorb the big shocks.

A solid steel ball one inch in diameter was observed to bounce twice to the height of about three feet on a hard floor and on the third bounce it fell on a rubber matting. When it struck the rubber matting it bounced about one inch and stopped dead. The resilience of the rubber matting absorbed the shock and killed the upthrow, the same as the "give" in the coils of the spring shock absorber kills the greatest part of the rebound of

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a car. If a man jumps from a table to the floor, alighting on his heels stiff-legged, it will give him a painful jar, but if he alights on his toes with knees bent, there is no discomfort whatever. The bend in his knees and the spring of the toes kill the shock and answer the same purpose as a spring shock absorber.

Recoil Shock Absorbers Required for Large Bumps

The large bumps, for which the recoil check type is required, do not occur oftener, on an average, than once in 800 feet, while in the space of 800 feet there are hundreds of small, jerky vibrations caused by cobble stones and railroad tracks to be eliminated by the cushion type of shock absorber. The cushion type, therefore, does more good than the recoil check type.

Each type, however, has its own particular use, depending upon the individual car to which it is applied. Several types of cars have springs which cannot be fitted with spring or cushion type shock absorbers, and such cars the recoil check will greatly help.

FRANK R. FARNHAM:—The development in passenger automobile spring suspensions has been due to two reasons: First, because sales have depended to a great extent upon the comfort of the passengers as well as upon the power, speed and flexibility of operation of the chassis. Second, it has been recognized by manufacturers of automobiles that the very vibration which it is desired to eliminate in order to secure the comfort of the passengers is also the cause of many minor repairs and adjustments to the engine, transmission and chassis in general.

Passenger automobile spring suspensions have been developed to a very high degree with a marked tendency toward the use of long and fairly flexible springs having a comparatively slow period of vibration. Yet even with spring suspensions of this character a large number of automobile owners have found it advisable to use shock-absorbing devices of one sort or another, in order to check the rebound of the springs when traveling over rough roads or pavements.

Since the total weights carried by most passenger automobiles are comparatively light, and since, in the interest of riding comfort, their spring suspensions have been developed to a very high degree, the effect of such spring stiffness under light loads is not intolerable.

Very different conditions, however, prevail in the case of the commercial vehicle.

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In designing motor truck springs the engineer has been forced by conditions to provide a suspension with an ample factor of safety, not only to carry the rated load, but also to cope with the excessive overloads which may be carried in emergency.

Such a factor of safety naturally presupposes a strong, rugged spring—one having ample strength to carry the truck chassis and the pay-load, and which under such load will show a fair degree of resiliency. This spring, however, will be deflected very little if any when the vehicle is operating empty or in the lightly loaded condition.

No one spring can be designed for motor truck service that will develop equal resiliency under both light and heavy loads, and this fact would indicate the advisability of using auxiliary springs which shall automatically come into action when the load on the truck is reduced to such a point that the main springs show insufficient resiliency. These auxiliary springs should also automatically go out of action and transfer their work to the main springs of the truck when the load carried is sufficient to develop the resiliency for which these were designed.

Such a dual type of spring suspension would effect a very material reduction in the vibration reaching the chassis when the vehicle is traveling lightly loaded or empty, and any vibrations reaching the chassis would be very much reduced in sharpness.

There is already on the market a device which actually does constitute dual spring suspension in connection with the regular leaf springs of a motor truck, such as above referred to. With such slight modifications as the design of a given truck may require, the Kim auxiliary spring consists of a pair of heavy coil springs of the torsion type. These are mounted in the yoke in such a way that their inner ends are attached to a riding member of the yoke, while the outer ends are fastened to a stationary member.

As long as a light load only is carried, or the truck is driven empty, this auxiliary arrangement performs practically all the spring action. When the truck is loaded up to or beyond its rated capacity, the torsion springs embodied in the Kim spring assume a locked position, in which they are subject to no further stress or action, and the entire load of the truck is carried on the regular leaf springs, which of course are then operating under conditions of maximum resiliency.

The entire assembly of the Kim auxiliary spring is so

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arranged that if through accident either or both of the torsion springs should be fractured the assembly would hold together, then performing the same service as the regular spring shackle which it replaces.

MR. CARTER:—I believe that a spring suspension of a given flexibility produces comfortable riding over only a narrow range of road, load and speed conditions. To produce the greatest comfort when riding at low speeds over cobbles and rippled roads very flexible springs should be used, while, conversely, for fast driving over heavy and rough roads stiffer springs are imperative, not only to secure moderate comfort, but also to adequately protect the car from damage.

Some automobile manufacturers attempt to meet these conditions by supplying springs of different flexibilities. One European manufacturer even goes so far as to have available springs of over thirty different flexibilities; the weight of the body, average number of passengers, conditions of roads encountered and the usual speed at which the car is to be driven are considered when the springs are selected for each car. Good results can thus be accomplished, as the roads are universally good in Europe, and those driving the cars have learned to go carefully over the few stretches of bad roads which may be encountered.

In this country, where road conditions vary so greatly, automobile manufacturers have been obliged to make compromises. Some use stiff springs to protect the car over the worst road conditions. Others use flexible springs requiring careful driving, but the majority use stiff springs on front and reasonably flexible springs on the rear.

Shock Absorbers Modify Spring Flexibility

The successful shock absorbers on the market can be divided into two classes:

1 Those acting as auxiliary springs, such as the Westinghouse, U. S. E., etc., which are attached between the end of the car springs and the frame. Their prime object is to add flexibility to the car springs.

2 Those which are mounted between the axles and frame and reduce the flexibility of the car springs in either one or both directions.

It is evident that where one type is successful the other type would be a failure. With the almost universal method of equipping American cars with stiff front springs and flexible rear springs, the Class 1 type of

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shock absorber should be used in front and Class 2 on the rear in order to obtain good results.

I have been attempting to make it practicable to equip cars on both front and rear with soft springs of high flexibility, allowing them practically free action under slight shocks such as on cobbles, rippled roads, and in city driving, while controlling their action by the use of an attachment called an oil spring which gives an increasingly greater resistance as the road condition becomes less favorable or as the load or speed may be increased.

The combined action of the oil springs and such a set of high flexibility springs gives in effect the action of flexible springs for good road conditions, and of increasingly stiffer springs as load and speed increase and the road surface becomes rougher.

It seems to be a mistaken idea that because the front springs are not under the passengers they do not affect riding. Nothing could be more erroneous, as all the disagreeable vibrations from too rigid front springs are transmitted through the car. Again, as the period of the stiff front springs is so out of phase with the more flexible rear springs, the result over certain roads is most unsatisfactory.

Suitable Spring Flexibility Important

When equipping cars of standard makes in this country with oil springs I have found that much better riding qualities can be obtained by increasing the flexibility of the front springs which, when operating in conjunction with the oil springs, do not break through and "hit bottom." From actual experience I have obtained exceptionally good results with springs of the following flexibilities when using oil springs to control their action.

Construction of Laminated Piston Oil Spring

The oil spring, Fig. 18, has a flexible laminated steel piston and a flexible collapsing spring steel curved partition operating within the oil chamber in order to obtain a graded resistance which produces but slight reaction against small movements of the axle and increasingly greater resistance the larger the magnitude of this movement becomes. It also produces a greater reaction against recoil at any one point than against impact.

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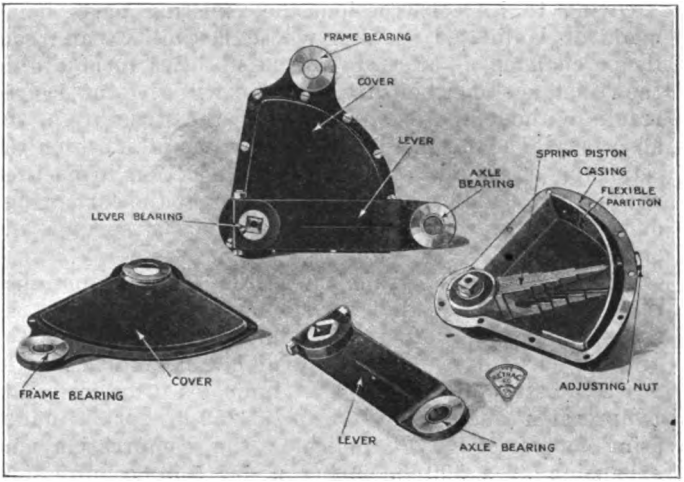

FIG. 18—INTERIOR CONSTRUCTION OF THE RETRAC OIL SPRING

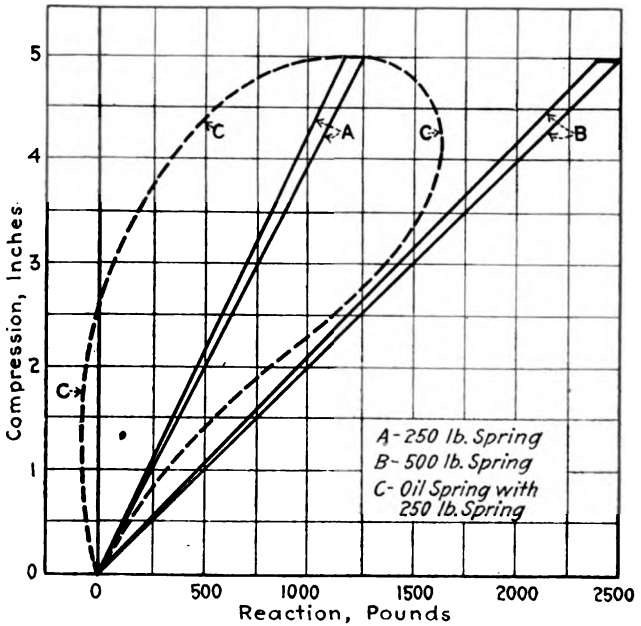

FIG. 19—SHOWING THE RESULTS OBTAINED BY COMBINING AN OIL SPRING WITH A MAIN SPRING OF HIGH FLEXIBILITY

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Owing to its construction, the resisting parts being made of laminated springs whose flexibility can readily be changed, it offers broad possibilities in obtaining the resistance in either direction that will give the desired results without in any way changing the structure of the absorber. The springs acting within the oil, cushion the retarding effect and allow a powerful action to be used without noticeably stiffening the action of the car springs or making the combined action of the car springs and absorber jerky.

Fig. 19 shows graphically the results obtained through the use of the oil spring combined with the action of a spring of high flexibility compared with the graphic curve of a stiff spring of low flexibility and of twice the strength per inch. *A* indicates graphically the compression and expansion of a 250-lb spring under 5-in. compression. *B* indicates the compression and expansion of a 500-lb. spring under 5-in. compression. *C* indicates the combined curve of the oil spring and the 250-lb. spring under similar conditions, based on a period of 130 vibrations per minute.

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STANDARDIZED SPECIFICATIONS FOR LUBRICATING OILS

BY C W STRATFORD*

In all industries the tendency toward standardization has never been stronger than at present. Under stress of war, international standardization committees are now the order of the day. The work done by these committees has been fruitful in securing the highest degree of quality, suitability and manufacturing efficiency. The oil industry appears to be one of the few remaining in which no effective standardization has been brought about. It is generally recognized that there is real need for some dependable method or methods of distinguishing between good and poor lubricating oils.

The advent of internal-combustion engines has introduced a new and important element into the lubrication of machinery, namely, much higher operating temperatures. Oils in these engines are not only exposed to high temperatures, but are also brought into intimate contact with carbureted air and products of partial or complete combustion, which contaminate them.

When the science of internal-combustion engine design was in its early stages little attention could be spared for the proper study of explosion-engine lubricants. As this type of engine approached perfection a difference in lubricating oils was soon perceived in service. Engine manufacturers soon found that certain oils gave good results while others were unsatisfactory. As their chief task, however, has been to produce engines, and not oils, their attention has been concentrated exclusively within their own field. In rare instances some engine manufacturers have made exhaustive experiments with lubricating oils, assuming that such oils bought in lithographed tins would always be of uniform quality if called by the same trade name. This assumption, erroneous on their part, has resulted in the complete nullification of much good that might have been derived from such engine tests. Unfortunately, intelligent tests cannot be

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made on lubricating oils in heat engines unless they are planned and carried out by operators who thoroughly understand all the features of internal-combustion engine operation, as well as the fundamentals of hydrocarbon chemistry and the important secrets of the oil-refining industry. Lubricating efficiency and oil quality are distinctly a chemical matter; hence any specification seeking to limit quality and to definitely fix suitability and uniformity must repose upon a strictly chemical or physico-chemical basis.

There are so many variables in the operation of internal-combustion engines that unless all tests are run under carefully studied and uniformly limited conditions no comparison can be made with the results obtained from the lubricants in practical use. Because of the difficulty and of the great expense attached to the execution of such tests, it is obvious that they should be supplanted by some laboratory methods which would make possible the determination of the same qualities and defects that are brought into unmistakable evidence in properly run engine tests or in actual service.

One of the first attempts to establish such laboratory tests was made by trying to apply the heat and the emulsion tests in the determination of the practical service value of lubricating oils. Oils that permanently emulsify with distilled water were found to decompose rapidly when used in an engine. Hydrocarbon oils showing a poor emulsion test also always suffer an undesirable decomposition when exposed to heat. Increased familiarity with these two tests soon proved to oil refiners that the best public demand was for engine oils that showed up most favorably under such tests. But the indications of these two tests were proved to be too limited to serve as a basis for judging all oils. When no "sulpho" compounds or highly unstable hydrocarbon compounds were present the results obtained were negative. Consequently other tests had to be evolved that would indisputably and definitely prescribe the stability (meaning resistance to chemical change) and the suitability (correct volatility for the operating conditions) of all oils for use in internal-combustion engines.

In the light of the most recent study and exhaustive experimentation, the oxidation test appears to be the only dependable and satisfactory method by which the stability of oils can be predicted when they are used in service. The main facts shown by this test are the evaporation loss and the rate at which solid hydrocarbons

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are formed by polymerization or precipitation when oils are exposed to the working conditions of an engine.

The distillation of finished oils under a high vacuum offers an unfailing means of determining their constituents; that is, of what groups of hydrocarbons, highly volatile or less volatile, the finished oils may consist. This test is a useful supplement to the oxidation test.

In the oxidation test oils are exposed to a uniform high temperature and the surface of the oil under test is swept by a continuous current of air, thus duplicating actual conditions to be met with in an engine.

Fractionation under conditions of high vacuum is an excellent method of making an approximate analysis of finished lubricating oils, since it permits of separating the volatile from the less volatile portions in graduated fractions. When oils show high evaporation loss the oxidation test alone cannot explain the reason for this loss. Evaporation may occur because of natural volatility or because of the occurrence of "cracking," that is, decomposition of the oil into light portions and heavy liquid or solid hydrocarbon residuals.

Customary Routine Tests

Before giving an extended description of the apparatus and the method of making the test it is, perhaps, well to discuss certain routine physical tests that have become traditional to the oil industry and to the oil consumer.

The problem of differentiation between groups of hydrocarbons sold under trade names is a different matter when viewed from the angle of the oil refiner, as compared with that of the final consumer. The oil refiner must have some practical tests that suffice for his purposes, but at the same time during the processing of his distillates, enable him to produce oils having uniform properties.

A perfect knowledge of his crude oil, as well as a familiarity with all of the distillates from that crude oil, automatically builds up for the refiner a series of standards which, though significant to him, are meaningless to the consumer. In most instances these standards are of little value to other oil refiners who are handling crudes of different characteristics. The meaning of these customary routine tests—gravity, flash, fire, viscosity, cold test and color—may be taken as examples. By these tests oil refiners accustomed to handling some specific crude or mixtures of crudes soon become so familiar with their distillates as to be able to condemn or approve any distillate in any stage of refining.

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Gravity Test—The first separation of crude oil into its distillates is usually controlled by their gravity. This test also indicates to the oil refiner the geographical source of the crude oil. Formerly, all oil refiners judged the quality of the light volatile products, such as fuels for engines, by gravity alone. Present-day developments, however, have shown that this is not a proper indication of volatile fuel characteristics. Gravity, as applied to any petroleum product, is of no practical significance to the consumer.

Flash Test—This is used by oil refiners as a measure of volatility. The very lightest products, such as gasoline, are not controlled by a flash determination. The flash point of all other distillates and residues is generally used as an indication for necessary separation of distillates into their required fractions. The flash test made on products not contaminated is an indication of the same property as shown by the fire test. When the products are contaminated the fire test shows more nearly the real constituents of the oil examined.

Neither flash nor fire tests are of major importance to the consumer because the real volatility of oils, as determined by distillation under high vacuum, gives him a true measure of oil economy in service.

Cold Test—The cold test is usually applied only to lubricating oil fractions, and is employed by the refiner to control the finishing of his products. It is of importance to the consumer in determining the suitability of oils for use under certain climatic conditions.

Viscosity Test—The viscosity of his products is of concern to the oil refiner only in so far as it may be required to meet the demands of the consumer. Through practical experience the consumer has learned what the correct viscosities of lubricants should be for different types of machinery.

Color Test—As the petroleum industry developed, certain products were gradually accepted as standard because of their most obvious characteristic, color. It has long been customary to include the color value of an oil as a necessary marketing requisite, along with its other properties. The refiner takes the color of oils into consideration as being a valuable indication of their uniformity, it being also a check on correct processing. Although in the past color was considered as an indication of purity, it has been demonstrated beyond any doubt that it alone means nothing in service to the ultimate user.

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The unusually rapid development of aviation engines, and their importance for war purposes, have been the direct causes for further investigation and the carrying out of intensive study of the relative qualities, special and otherwise, of all available lubricating oils in the American market intended for use in automobile, truck and aircraft engines. In continuing this study, naturally, due consideration was given to routine oil testing as now made by oil refiners. Of these tests, gravity, flash, fire and color were soon eliminated as being non-essential, because they gave no decisive indication of the decomposition of an oil when exposed to heat. The viscosity, cold test, volatility, susceptibility to oxidation, and emulsifying properties were substituted as being the only tests actually required. These five tests have been found to prescribe completely and accurately the stability and suitability of lubricants for the above-mentioned engines.

Recommended Lubricating Oil Specifications

The following specifications, which are recommended by the author for lubricating oils to be used in internal-combustion engines, are based on the five tests determined to be essential. In order to put them into force it would be absolutely necessary to standardize the apparatus used in the determination of all the properties indicated. This also applies to the standardization of petroleum ether. Given below are a description of this apparatus and the specifications of the petroleum ether, both of which are recommended for standardization.

The nomenclature used for designating the specifications is as follows: Aircraft engine oils, A. E. O.; automobile and truck engine oils, A. T. O.; and Knight sleeve-valve engine oils, K. S. V. O. The air-cooled motorcycle engine oils are not given a separate designation, since the oils specified for other duty can be used.

AIRCRAFT ENGINES

A. E. O. No. 1 (*High Altitude or Low Altitude Winter*)

1 Viscosity at 212 deg. fahr., seconds . 58 to 70; desired 65

2 Cold test . . . . . Maximum 40 deg. fahr.

3 Boiling-point range,
Maximum 4 per cent under 300 deg. cent.

4 Oxidation . . . . . Insolubles, maximum 0.20 per cent
Evaporated loss, maximum 40 per cent

5 Emulsion . . . . . Maximum trace

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A. E. O. No. 2 (Low Altitude Summer)

1 Viscosity at 212 deg. fahr., seconds .80 to 90; desired 85
2 Cold test.....Maximum 45 deg. fahr.
3 Boiling-point range,
Maximum 4 per cent under 300 deg. cent.
4 Oxidation.....Insolubles, maximum 0.20 per cent
Evaporation loss, maximum 40 per cent
5 Emulsion .....Maximum trace

AUTOMOBILE AND TRUCK ENGINES

A. T. O. No. 1 (Summer)

1 Viscosity at 212 deg. fahr., seconds .52 to 60; desired 55
2 Cold test.....maximum 35 deg. fahr.
3 Boiling-point range,
Maximum 30 per cent under 300 deg. cent.
4 Oxidation.....Insolubles, maximum 0.30 per cent
Evaporation loss, maximum 40 per cent
5 Emulsion .....Maximum trace

A. T. O. No. 2 (Winter)

1 Viscosity at 212 deg. fahr., seconds .45 to 50; desired 48
2 Cold test.....Maximum 30 deg. fahr.
3 Boiling-point range,
Maximum 30 per cent under 300 deg. cent
4 Oxidation.....Insolubles, maximum 0.30 per cent
Evaporation loss, maximum 50 per cent
5 Emulsion .....Maximum trace

KNIGHT SLEEVE-VALVE OIL

K. S. V. O. (Summer)

1 Viscosity at 212 deg. fahr., sec. 100 to 120; desired 115
2 Cold test.....Maximum 52 deg. fahr.
3 Boiling-point range,
Maximum 3 per cent under 300 deg. cent.
4 Oxidation.....Insolubles, maximum 0.20 per cent
Evaporation loss, maximum 30 per cent
5 Emulsion .....Maximum trace
Winter, use A. E. O. No. 2

AIR-COOLED MOTORCYCLE ENGINES

Winter—One and two-cylinder types, use A. E. O. No. 2
Four-cylinder types, use A. T. O. No. 1
Summer—One and two-cylinder types, use K. S. V. O.
Four-cylinder types, use A. E. O. No. 2

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I Viscosity

It will be necessary to determine only the viscosity at 212 deg. fahr. For the purposes of a more thorough investigation the rate of fall of viscosity between low and high temperatures, three points may be found: 100, 150 and 212 deg. fahr.

Care should be exercised in maintaining the temperature of the oil being run at 212 deg. fahr.

The instrument used will be a standard Saybolt universal viscosimeter.

Limit of error plus or minus one second.

II Cold Test

The cold-test box (18 by 18 by 14 in. high) is filled about three-quarters full with ice and salt mixture (about 10 to 1 by volume). Copper cylinders 9 in. high, and having an inside diameter of 2 in., are inserted in this mixture so that the tops of the cylinders are about level with the surface of the ice and salt mixture. Regular 4-oz. oil sample bottles and approximately 5-in. immersion, cold-test thermometers, having a range of from 20 to 140 deg. fahr., complete the equipment. The procedure of test is as follows:.

A sample bottle is filled about one-third full of oil, and a cork carrying the thermometer is inserted in the neck of the bottle so that the bulb of the thermometer is immersed in the oil. The sample is then placed in one of the copper cylinders. The bottle should not be allowed to come in contact with the bottom of the tube, a thin section of cork being interposed between the two. The temperature of the oil, as shown by the thermometer, is carefully observed, and at each drop of 2½ deg. fahr. the sample is removed and examined. The temperature at which the oil just refuses to flow when the bottle is held in a horizontal position is taken as the cold test.

The above method is for oils having a cold test under 40 deg. fahr. Those having cold tests above this point are tested as follows:

The point at which the oil just refuses to flow is obtained in exactly the same manner as described above, and the oil is then chilled 20 deg. fahr. below this point. The sample is then removed and placed in water which is kept just 5 deg. warmer than the oil, as shown by the thermometer in the bottle. The sample is tested at each 2½ deg. fahr. increase in temperature, and the temperature at which the oil will flow evenly across the bottle

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when it is held in a horizontal position is called the cold test.

The limit of error is plus or minus 2 deg. fahr.

Annapolis Pour Test—The procedure is exactly the same as that given above for oils under 40 deg. fahr. cold test, except that 5 deg. fahr. are added to the point observed as cold test and reported as pour test.

PROPERTIES OF REPRESENTATIVE AMERICAN LUBRICATING OILS FOR USE IN INTERNAL COMBUSTION ENGINES

Kind of Oil	Physical Properties Fresh Sample						Vacuum Distillation	Oxidation Oven	
	Baumé Grav.	Flash, Deg. F Open Cup	Deg. F Burn	Deg. F Chill	Viscosity, Seconds 100° F	Viscosity, Seconds 212° F	% Dist. Under 300° C	% Evap. Loss	% Insol.
Aerial Oil	29.1	425–	460	50	563	71	13	18.98	0.0940
Amalie Special	31.0	430	460	40	309	43	4	21.71	0.0230
Atlas Aerial Heavy	27.8	456	540	47	506	55	328° I.B.P.	17.84	0.0130
Cadillac Detroit Lt	28.2	405	470	11	190	..	44	..	..
Duplex No. 850	30.7	425	495	27	194	46	1	21.49	0.0595
Havoline									
Light	25.9	370	430	33	173	42	22	62.98	0.5800
Medium	25.0	385	450	34	227	46	6	57.74	0.8600
Heavy	25.6	395	455	46	361	54	18	40.49	0.1500
Mobiloil									
Zeta Light	26.0	375	430	31	141	42	28	51.75	0.3010
Zeta Medium	24.2	400	470	42	265	47	29	35.06	0.3300
Zeta Heavy	25.6	420	480	42	476	62	301° I.B.P.	27.64	0.0404
E Light	26.1	370	430	0 ◉ 0	167	44	21	76.44	0.1962
A Medium	21.8	360	430–	24	230	49	21	57.06	0.6381
B Heavy	28.2	500	550	41	1640	122	2	25.26	0.1756
Arctic Lt. Med	29.3	370	425	6	221	45	28	71.53	0.3093
Arctic Medium	21.1	370	430	8	300	46	19	49.06	0.4160
BB Med. Hvy	25.8	460	540	46	936	86	344° I.B.P.	18.67	0.0290
Monogram									
Light	27.6	360	410	20	140	41	14	67.12	0.4600
Medium	26.0	375	430	23	289	50	16	46.19	0.1800
Heavy	28.9	430	505	34	340	55	306° I.B.P.	25.75	0.0700
Ex. Hvy	24.7	465	555	58	1583	110	3	13.10	0.0440
Oilzum									
Heavy	29.1	430	500	30	361	50	310° I.B.P.	15.86	0.1147
Crystal G. E. Oil	17.0	505	590	2	1060	83	.....	17.84	....
Perfection									
A Light	29.1	400	470	26	181	45	314° I.B.P.	49.36	0.1378
B Medium	24.9	390	450	22	243	47	2	32.23	0.3704
C Heavy	29.3	420	495	40	316	54	2	27.53	0.0088
Polarine, New Jersey	24.6	395	470	6	221	43	57	48.78	0.6300
Quaker State Medium	28.8	440	520	21	301	48	29	23.43	0.0540
Service Auto Oil	24.3	390–	460	33	195	44	7	38 91	0.9600
Socony									
Zero	24.2	395	470	St. ◉ 0	219	46	3	42.85	0.1531
Polarine Hvy	25.4	385	453	35	300	51	7	38.61	0.1358
Supreme									
Light	22 3	325	360	St. ◉ 0	137	39	72	60.21	0.7689
Medium	19.8	330	380	St. ◉ 0	217	43	49	56.37	0.7041
Heavy	19.4	330	370	0 ◉ 0	342	44	42	53.37	0.4609
Texaco									
Light	21.3	325	380	St. ◉ 0	208	42	44	59.13	0.1285
Medium	20.9	350	400	St. ◉ 0	301	46	21	51.40	0.1000
Heavy	19.3	355	420	10	495	51	19	43.87	0.0874
Urus	19.7	390	455	St. ◉ 0	735	58		37.63	0.2787
Valvoline Medium	30.7	385	440	30	267	51	29	38.85	0.0250
Veedol									
Aero No. 1	26.2	455	535	39	795	78	3	34.96	0.3956
Aero No. 2	27.1	450	530	38	614	80	3	36.58	0.1614
Aero No. 3	26.3	435	530	38	517	68	318° I.B.P.	27.96	0.0388
Aero No. 4	27.6	440	515	34	513	64	3	28.11	0.0416
Aero No. 5	24.7	440	530	38	418	55	326° I.B.P.	28.83	0.0741
Aero No. 6	24.7	450	540	37	474	58	318° I.B.P.	18.91	0.0722
Zero Light	26.1	390	460	14	199	44	57	39.56	0.4545
Medium	26.3	405	475	20	250	48	23	49.54	0.0301
Heavy	26.2	410	480	33	329	52	26	30.08	0.0650
Extra Heavy	26.1	465	550	44	..	111	3	9.68	0.3664
Waverly	31.0	400–	460	24	160	40	37	44.76	0.1400
White Star Motor Oil	25.8	360	415	18	170	43	17	59.65	0.4800
Wolf's Head									
Light	31.2	390	445	34	142	40	1	56.70	0.1750
Medium	28.8	410	470	26	223	46	310° I.B.P.	45.05	0.3800
Heavy	28.6	415	475	32	324	52	304° I.B.P.	35.50	0.2100
No. 8	27.6	465	550	46	1196	100	3	38.83	0.0470

The above method is used for all oils except black oils. The cold test for black oils is taken at the Annapolis Experimental Station as follows:

The oil is chilled by exactly the same procedure as above; when the oil just refuses to flow the sample is

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removed, the bottle held horizontally, and the oil stirred with the thermometer. The temperature when the oil flows freely is recorded as the pour test.

A. S. T. M. Pour Test—The method specified by the A. S. T. M. is exactly the same as that used at the Annapolis Station, except that the sample should be removed and tested for every 5 deg. drop in temperature, instead of 2½ deg. fahr., as used at Annapolis. The difference between the A. S. T. M. and Annapolis methods is about 2 deg., and generally the Annapolis method will give the higher test.

III Boiling-Point Range (Vacuum Distillation)

The apparatus for making this test consists of a special steel still and vapor tower, a special water-cooled condenser and a chamber for the distillate receivers. The position of these receivers can be adjusted from the outside without alteration of the pressure in the distillation system. The distillation system is connected with a pump capable of maintaining a vacuum of 2½ mm. absolute and having a capacity of 6 cu. ft. per minute.

The still is charged with 2000 cc. of the oil under examination. The standard vacuum, 40 to 50 mm. absolute, is established throughout the closed system, and heat applied to the bottom of the still. For the purpose of the specification it will be necessary to carry the distillation only to the point at which the still temperature has reached 300 deg. cent. (572 deg. fahr.). The percentage of distillate up to 300 deg. cent. has been arbitrarily taken as a basis for judging the relative volatility of different lubricating oils. However, for the ultimate analysis of finished lubricating oils, the distillation may be continued up to the point where nine-tenths of the total charge will have passed over, the rest being allowed to remain in the still, and estimated as residual.

The fractionation of finished lubricating oils under high vacuum is a satisfactory way of determining boiling-point limits with the least decomposition. The quantity of low-boiling-point fraction varies widely in lubricating oils intended for use in different types of internal-combustion engines. All engines that operate under approximately full load and at high temperatures (aviation engines) should contain a minimum quantity of low-boiling-point fractions, say 4 per cent distilling under 300 deg. cent. The results obtained by actual tests in aircraft engines clearly demonstrate this point. On the other hand, passing from such conditions of high

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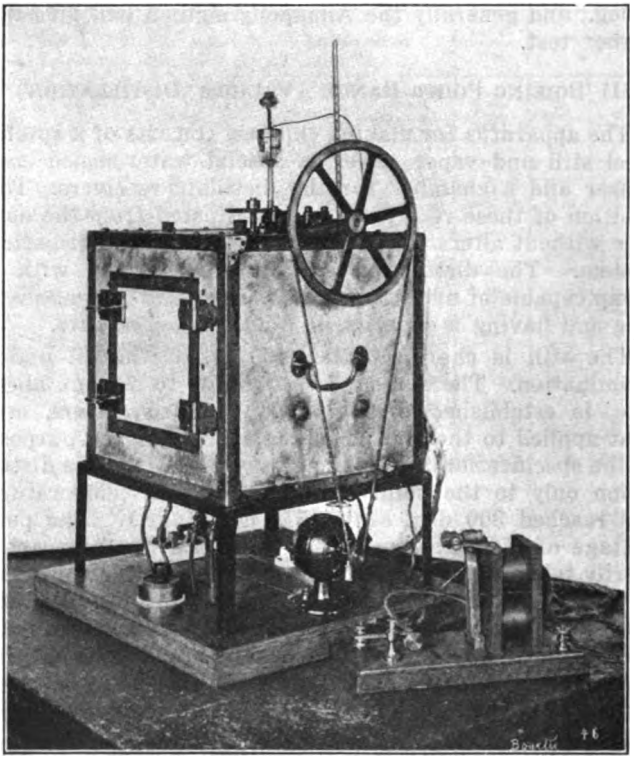
operating temperatures to those of widely varying and comparatively low operating temperatures (passenger cars and trucks), it is necessary to increase materially the quantity of low-boiling-point fractions to prevent serious carbonization. There is a limit, however, to the addition of low-boiling-point fractions, for the reason that too large a quantity will greatly increase the specific consumption of the lubricant in service; hence some



OXIDATION OVEN OF MODIFIED WATERS-TYPE

mean value must be fixed upon, a permissible balance being struck between carbonization and high consumption. An analysis of the most successful automobile oils now sold shows that not over 30 per cent of the oil should distill below 300 deg. cent. in the vacuum distillation apparatus described. This volatility is a guarantee of a reasonably low specific consumption, good lubrication and minimum carbonization within the explosion chambers.

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Oxidation Test

The apparatus for making the oxidation test consists of a cubical box constructed of heat-insulating material. Through the vertical axis of this box passes a shaft, at the center of which is attached a disk of about 11 in. diameter. This disk and shaft are rotated at a speed of 25 r.p.m. The containers for oil samples are steel cups cut from the solid, 50 mm. inside diameter, plus or minus 0.04 mm.; and 50 mm. high, plus or minus 0.4 mm. The cup wall is 1 mm. thick. The oven is electrically heated by resistance coils, and the temperature is maintained constant and regulated by a thermostat. Air circulation is provided through holes in the bottom of the sides of the oven and through the top. The determinations made in this oven are:

1 Evaporation loss.

2 Petroleum ether insolubles.

3 Varnish.

Fifty grams of the oil are weighed into the cups, and these are placed at equal distances on the outer edge of the rotary disk. The temperature is brought to 225 deg. cent., and the test is continued at this uniform temperature for twelve hours, in two periods of six hours. The samples are then removed and allowed to cool, weighed, and the loss in weight (per cent) recorded as evaporation loss. The contents of the cups are then washed through a Gooch crucible with standardized petroleum ether (see below) until the filtrate is colorless. From the crucibles on which remain the insoluble residue the petroleum ether is allowed to evaporate, after which they are introduced into a Freas oven and thoroughly dried at a temperature of 105 deg. cent. The weight of the residue is then determined and the percentage recorded as petroleum ether insolubles.

When the metal cups have been thoroughly washed with petroleum ether there still remains a varnishlike residue, which clings to the upper walls. The weight of this substance is determined and recorded as varnish.

The real significance of varnish values has not been clearly established up to the present. Poorly refined oils, however, as well as those from some mid-continent and from California crudes, usually show high varnish values.

The maximum variation of checked results on evaporation loss and petroleum ether insolubles should not exceed 2 per cent.

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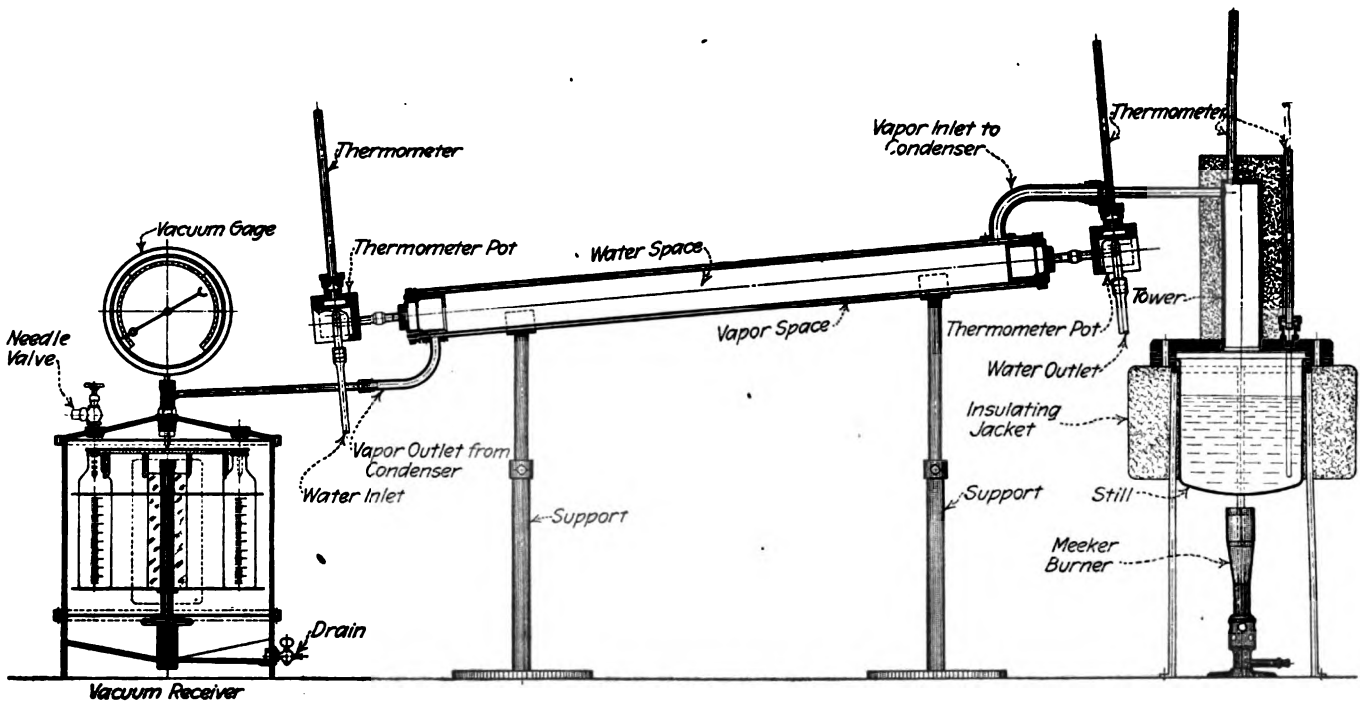

CROSS-SECTION OF VACCUM DISTILLATION APPARATUS SHOWING RECEIVER, STILL AND OTHER PARTS

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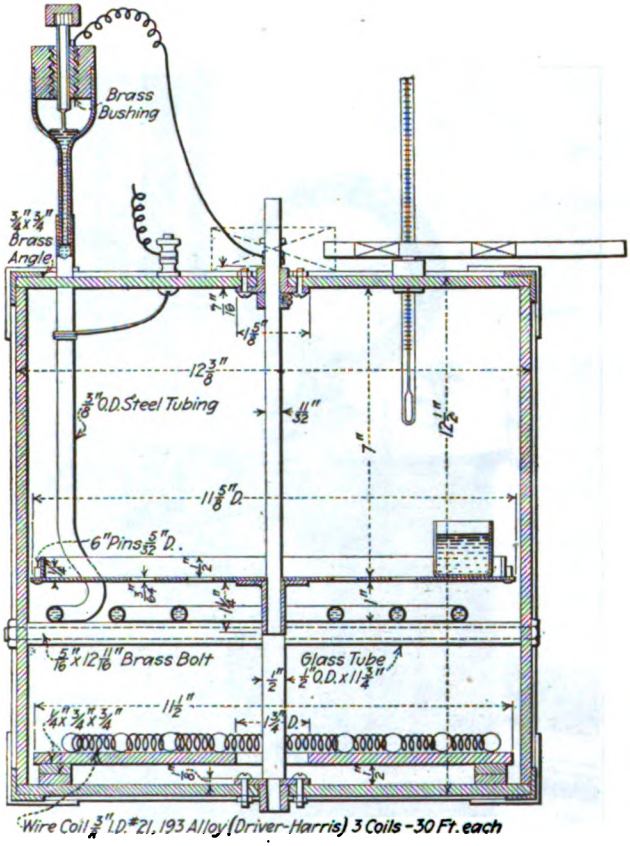
It has been found that the evaporation loss of an oil does not always coincide with its boiling-point range; therefore, it seems permissible to conclude that the evaporation loss is a measure of the decomposition of an oil,


Brass Bushing
Brass Angle
3/8" O.D. Steel Tubing
6" Pins 5/32" D.
5/16" x 12 11/16" Brass Bolt
Glass Tube
1/2" O.D. x 11 3/4"
Wire Coil 3/8" I.D. #21, 193 Alloy (Driver-Harris) 3 Coils - 30 Ft. each

CROSS-SECTION OF OXIDATION OVEN

as well as of its volatility. In judging oils it is important to observe carefully the relation between the boiling-point range and the evaporation loss. For example, an oil may have a high boiling-point range, and yet, owing to its instability, may readily decompose when exposed to heat, and thus show a high evaporation loss.

The petroleum ether insolubles indicate, on the one hand, the solid carbonaceous matter formed in an oil, which splits up when exposed to heat into light volatile

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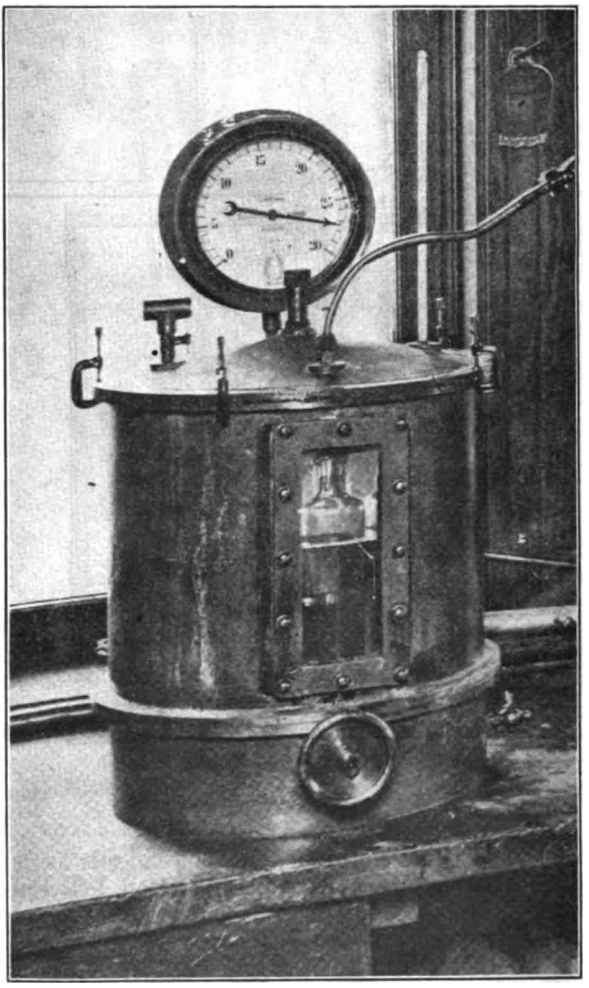
ends and solid products. On the other hand, these insolubles may indicate the precipitation of colloidal carbon or other products in improperly refined oils. Values of petroleum ether insolubles may be entirely independent of those of boiling-point range or evaporation loss.



RECEIVER FOR DISTILLATION APPARATUS

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Standardized Petroleum Ether

For the determinations of petroleum ether insolubles it is imperative that a standardized petroleum ether be employed, such as the following:

Petroleum ether should be made from steam-stilled Prime City naphtha from Pennsylvania crude, which has the following properties:

Gravity, deg. Baumé, 72 to 76.

Initial boiling point, deg. fahr., 110 to 122.

Final boiling point, deg. fahr., not over 270.

Petroleum ether should conform to the following specifications:

Gravity, deg. Baumé, 77 to 80 (the matter of gravity is unimportant).

Initial boiling point, deg. fahr., 110 to 120.

Final boiling point, deg. fahr., 220 to 245.

Unsaturated content not over 4 per cent by absorption in sulphuric acid (Bureau of Mines method).

The Saybolt distillation method applies to both of the above boiling point ranges.

V Emulsion Tests

In making emulsion tests the simple apparatus described below gives excellent check results.

The apparatus consists of a Coles milk shaker slightly altered to receive two 4-oz. oil sample bottles. The shaker is operated at a speed of 600 r.p.m., and has a 2-in. stroke.

About 1½ in. of oil and an equal quantity of distilled water is poured into the bottles, which are then corked and placed in the machine. These are shaken for half an hour, after which they are removed and allowed to stand at room temperature for 24 hours. The amount of unseparated oil is then determined and recorded as emulsion.

The above method applies only to oils having a viscosity not exceeding 50 sec. at 212 deg. fahr. When the viscosity exceeds this figure the oil should be blended with 50 per cent of standardized petroleum ether before shaking.

For the purpose of the specification the emulsion value of an oil can be considered as a quantitative test for "sulpho" compounds.

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